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THE "COMPLETE ENGINEER" SERIES • VOLUME 12

MECHANICAL POWER TRANSMISSION

SPECIALLY INTENDED FOR FACTORY ENGINEERS, FITTERS, ERECTORS, AND ALL CONCERNED WITH INDUSTRIAL POWER TRANSMISSION

Prepared by a Staff of Technical Experts under the direction of

E. MOLLOY

WITH ONE HUNDRED ILLUSTRATIONS

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THE “COMPLETE ENGINEER” SERIES

VOL.
1 INTRODUCTION TO WORKS PRACTICE
2 AIRCRAFT PRODUCTION
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PREFACE

THE subject with which this book deals is one of great importance to the engineer, as upon the efficiency and smoothness with which power is transmitted from the prime mover to the driven machine depends the efficient operation of countless factories and industrial undertakings.

Among the mechanical units used in power transmission are included belts, ropes, pulleys, shafting, couplings, and clutches, and chapters are devoted to descriptions of the methods of installing and maintaining these items. The data contained in the first chapter includes notes on size of shafting to transmit a given horse-power, the use of collars and couplings, both of the compression and flexible type, together with dimensions and horse-power tables for couplings. Ball and roller bearings are also discussed, complete instructions being given for assembly and mounting.

Some useful notes on the fitting of belts are given in the second chapter, particular attention being devoted to the methods of jointing and splicing belts, both of the balata and leather type. The illustrations show clearly how the operations are carried out using various types of belt fastenings. A noteworthy feature is the summary of troubles likely to be experienced with belt drives, together with suggestions for their remedy. A separate chapter contains notes on V-rope drive, including instructions for calculating horse-power transmitted and advice on suitable rope speeds.

Clutches are dealt with in two separate chapters, the first concerning purely mechanical types of clutch, while the second is devoted to magnetic clutches. The former chapter, after giving horse-power tables for the various types of clutches, includes full particulars of the fitting and maintenance of cone clutches, internal expanding clutches, band and rim clutches, and multiple disc clutches. In the section devoted to the operation of clutches, the use of both compressed air and electricity is referred to.

E. M.

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MECHANICAL POWER TRANSMISSION

Chapter I

GENERAL CONSIDERATIONS

IMPORTANT changes have taken place recently in the method of transmitting power from prime mover to machine.

The purely mechanical method which may be considered as including shafting, gear-wheels, chains, ropes, belts, etc., has now a serious rival in electricity.

In some instances electrical driving may possess advantages. On the other hand, depending on the lay-out, cost of current, etc., it may be that mechanical transmission is to be preferred. It cannot be stated definitely that one or the other system should be adopted in all cases.

In actual practice it is usually found that a combination of the two systems is necessary to fulfil the requirements of economical workshop driving.

The possibilities of electrical equipment are enormous and its suitability is almost unlimited, but to have a large number of electric motors is not always the cheapest way of driving.

In deciding the relative merits of the two methods for any particular installation, running and maintenance expenses should be considered as well as first cost, and, in addition, convenience of control must not be overlooked.

Factory Act Requirements

The new Factory Act in relation to Power Transmission Machinery states that " *Efficient Devices or Appliances should be provided and maintained in every room or place where work is carried on, by which the power can be promptly cut off from the transmission machinery in that room or place.*"

This control of power can be accomplished by the installation of separate motors for each room, or by the introduction of friction clutches as described in the previous article.

The most important condition to be fulfilled by any transmission system is that there shall be no involuntary stoppages of driven machinery, and in this respect the power transmission materials are as important as the prime mover or driven machinery, yet seldom are they given the same consideration, either in first choice or in maintenance.

It is, of course, necessary that a power-generating unit shall be relatively large in order to produce power at an economical cost, but machines, such as comprise the load in modern mills, and workshops are relatively

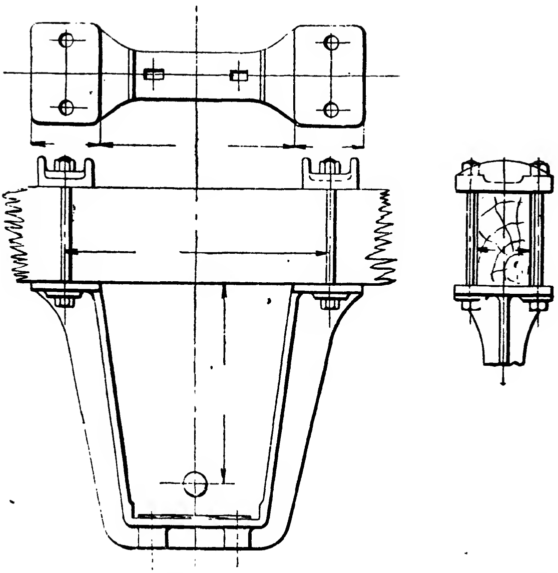


Fig. 1.—Sling hanger fixed to a narrow wood beam at right angles to shaft

small. The problem, therefore, is to drive at a reasonable cost a number of small machines, each requiring only a fraction of the power which can be generated cheaply.

There will be many drives where small-unit motors are desirable, but it will often be found possible to drive several machines from a single large heavily loaded motor or other prime mover, and if this method is adopted, it can be counted upon to reduce very substantially the investment and maintenance cost per installed horsepower compared with individual motor drives.

Arrangement of Machinery

In arranging machinery, the best plan to adopt is usually to place the heavy machines as near as possible to the prime mover, and drive them by wheel-gearing, ropes, or belts, according to the power required and distance from power house.

It is usual to drive heavy machines by gearing or ropes in conjunction with suitable friction clutches; lighter machinery is better driven by belts in conjunction with fast and loose pulleys, friction clutches, etc., or alternatively by individual electric motors.

Unnecessary duplication of belt or rope drives, countershafting, etc., is to be avoided, since every shaft and bearings absorbs a certain amount of power; hence if one section of the works must be running when another is stopped, it is advisable to arrange that all the shafting in the latter section is brought to rest.

When arranging shafting, etc., see that the supports are rigid and capable of maintaining it in correct level and alignment.

Roof-beams, stanchions, etc., should not be drilled, but the bearing soleplates, brackets, etc., should be arranged to bolt round them (Fig. 1).

Arrange bearings, clutches, gear-boxes, etc., in such positions that lubrication can be attended to without difficulty.

If these parts are placed in inaccessible positions they are often neglected.

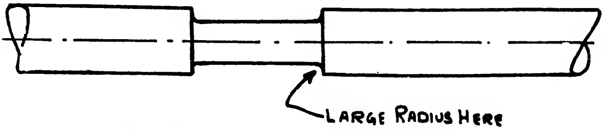


Fig. 2.—DIAGRAM ILLUSTRATING REFERENCE TO REDUCTION OF STRENGTH ACCOMPANYING REDUCED DIAMETER TURNED OUT OF A SHAFT

Shafting

Shafting may be cold-rolled or turned ; it is generally considered amongst metal-workers that turned shafting is the more desirable.

The method of manufacture of cold-rolled shaftings is such that a crystallisation of the surface is produced, and whilst it is contended that this process raises the elastic strength, it follows that, when the surface is penetrated for purpose of keyways, etc., the cold-rolled shell or skin springs and introduces difficulties in restraightening.

Shafting as used in millwright work not exceeding 5 in. diameter is produced almost entirely from round hot-rolled bars having an ultimate tensile strength of 28 tons to 32 tons per square inch, with an elongation of 20 per cent. in 8 in.

Larger shafts and shafts with collars, etc., are turned from steel forgings.

A reduced neck or sunk journal turned out of a shaft (*see* Fig. 2) causes a serious loss of strength and stiffness and should be used only after the stress to which the shaft is to be subjected has been carefully considered ; further, there should be no sharp corners at the position where the shaft is reduced in diameter (*see* Fig. 2). This is a very important point and should not be overlooked.

Where possible shaft lengths should be arranged to be less than 24 ft., as the English railway companies make extra carriage charges for this length unless in two-ton lots.

Diameter of Shafting to transmit a Given Power

To determine the diameter of shafting required to transmit a given power, it is necessary to take into account :

(1) Speed.
(2) Distance between supports.
(3) Weight of pulleys, gear-wheels, etc.
(4) Pull of belts, thrust of gears, and angle of same.
(5) Nature of load.

There are many tables published which give the horse-power that plain mild-steel shafting of good quality should transmit, but as these are all based on torsional strength, they are best used as a guide only—for preliminary approximation—unless the weight supported, together with the pull due to belts, etc., is small compared with the size of shaft.

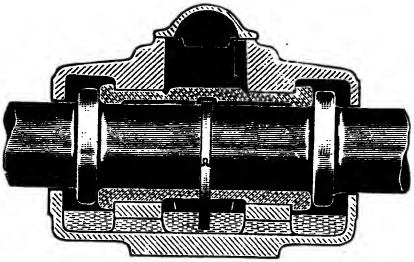


Fig. 2A.—FORGED COLLAR NECK ON SHAFT TO TAKE END THRUST

Shafting running at speeds higher than those usually encountered in power-transmission work requires special consideration, but for average conditions the following well-known formulæ are useful.

If a shaft carries a pulley, as in Fig. 3, of radius R over which a band or belt passes having a tension $T1$ and $T2$ lb., so that the effective turning force at the rim is $T1 - T2 = P$ lb., then the product of $P \times R$ is the twisting moment or torque, and is usually expressed in pounds and inches.

Notation

N = Speed of shaft in revolutions per minute.
H = Horse-power transmitted.
P = Force in pounds acting at a perpendicular distance of R inches from centre of shaft.
T = Torque or twisting moment on shaft in inch pounds = $P \times R$.
d = Diameter of solid shaft.
fs = Shear stress.
B = Maximum bending moment in inch pounds.
Te = Equivalent twisting moment due to combined twisting and bending.

$$H = \frac{2 \times 3{\cdot}1416 \times T \times N}{12 \times 33{,}000} = {\cdot}00001587\, TN \quad \ldots\ldots \quad (1)$$

$$\text{and } T = \frac{12 \times 33{,}000 H}{2 \times 3{\cdot}1416 N} = \frac{63025 H}{N} \quad \ldots\ldots \quad (2)$$

For round solid shafts :

$$T = \frac{3{\cdot}1416\, d^3}{16} fs = {\cdot}19635\, d^3 fs \quad \ldots\ldots \quad (3)$$

$$d = \sqrt[3]{\frac{16\, T}{3{\cdot}1416\, fs}} = 1{\cdot}72 \sqrt[3]{\frac{T}{fs}} \quad \ldots\ldots \quad (4)$$

When the shaft is subjected to bending in addition to torsion, the equivalent twisting moment can be found thus :

If Te = equivalent twisting moment, then $Te = \sqrt{(B^2 + T^2)}$ (Guest Max. Shear Theory) . . . . (5)

If the above value of Te is used in Formula (4), $d = 1{\cdot}72 \sqrt[3]{\frac{Te}{fs}}$ (6)

The diameter, speeds, transverse strains, and position of loads on shaft are the factors which determine the safe distances apart of shaft bearings; for average conditions, the tables published by Messrs. Crofts (Engineers) Ltd., and reproduced in Data Sheet No. 1, may be of service.

For line shafting the *fs* may be increased to about 4,800 lb. per square inch; therefore the horse-power given in the table in Data Sheet No. 1 may be increased by 50 per cent.

For shafting which does not carry any pulleys *fs* may be increased to about 6,400 lb. per square inch; therefore the horse-power given in the table may be increased by 100 per cent.

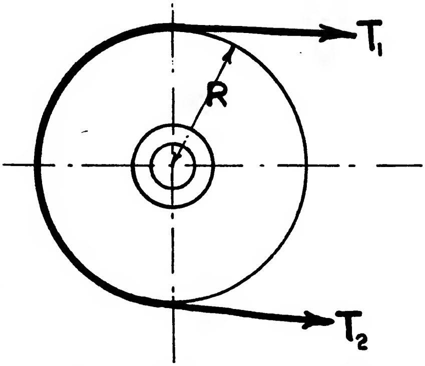


Fig. 3.—Shaft carrying pulley

Collars

All shafts require to be located in the bearings, and this is done by collars, the actual design of which depends upon the thrust to which they are subjected. For heavy thrusts sunken journals (Fig. 2) or forged collars (Fig. 2A) are used.

For installations where there is no definite end thrust loose collars (Fig. 4) are permissible; they facilitate erection as well as being cheaper than forged collars.

On account of expansion a shaft should be located by collars in one bearing only.

A shaft 60 ft. long will increase in length by about $\frac{1}{8}$ in., with a rise in temperature of 25° F.; it is, therefore, absolutely essential to make proper provision for this increase in length, and expansion couplings, usually of the claw type, are used (Fig. 5).

Couplings for Shafting

Couplings should always be placed close to a bearing, and when they are of the rigid type, as in Fig. 6, they should be forced into position on the shaft either by hydraulic pressure or screw press.

After keying they should be turned up true in the lathe. If removed after this operation, they are liable to be thrown

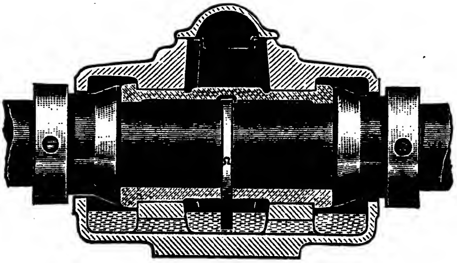


Fig. 4.—Loose collar

This type of collar is used for installations where there is no definite end thrust.

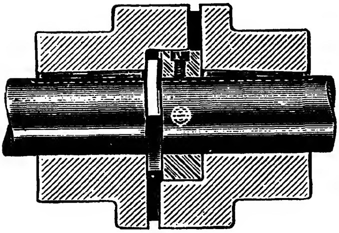


Fig. 5.—Expansion coupling

This is constructed to make provision for expansion and contraction of lineshafts and at the same time to retain concentricity.

out of truth by replacement and rekeying.

This difficulty led to the adoption of the split-sleeve type (Fig. 7) as well as various designs of compression couplings which drive due to friction "nip" on the shaft.

These designs permit shafts being delivered apart from the couplings, thus minimising the risk of bending in transit, and enabling long slender bars to be bundled and boxed together for transit.

Description of Compression Coupling (Fig. 8)

A double-coned slotted sleeve compressed on to the shafts to be coupled by means of two correspondingly coned safety flanges, which are drawn over the sleeve by bolts.

In the sleeve are six slots, five of which are sectional, the sixth being

Diameter of Bore . .	in.	1	1¼	1½	1¾	2	2¼
Length Overall, Standard Type . . .	in.	3⅛	3⅝	4⅛	4⅝	5⅛	5¾
Diameter of Bore . .	in.	2½	2¾	3	3¼	3½	4
Length Overall, Standard Type . . .	in.	6¼	6¾	7¼	7½	7¾	8¾
Diameter of Bore . .	in.	4½	5	5½	6		
Length Overall, Standard Type . . .	in.	9⅞	10⅞	11⅞	12⅞		

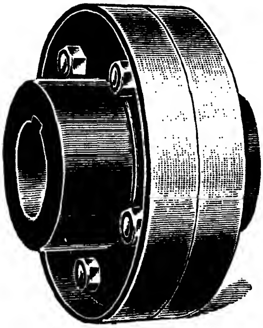


Fig. 6.—Rigid shaft coupling (pulley type)

Diameter of Bore .	in.	1½	1¾	2	2¼	2½
Diameter Overall .	in.	4½	5	5¼	6¼	6¾
Length Overall .	in.	5⅛	5½	5⅞	6⅝	7¼
Diameter of Bore .	in.	2¾	3	3¼	3½	4
Diameter Overall .	in.	7	7½	8¼	9	9¾
Length Overall .	in.	7⅝	8	12	13¼	15
Diameter of Bore .	in.	4½	5	5½	6	
Diameter Overall .	in.	10½	12	13	15½	
Length Overall .	in.	15¼	18	20¼		

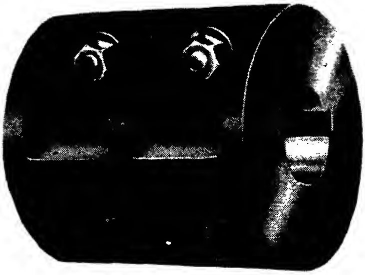


Fig. 7.—Split sleeve couplings

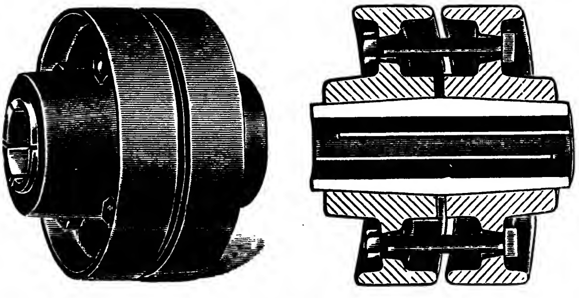


Fig. 8.—Safety flange compression couplings

cut right through the full length, thus ensuring a uniform and powerful grip, although the shaft may be slightly under-size.

Fixing Instructions

Before fixing see that both shaft ends are truly round, straight, and free from burrs. Make sure that neither shaft enters beyond the centre of the sleeve.

Draw the outer shells together over the sleeve and see that alternate bolts are gradually tightened one after the other.

Tapping the outer end of the hubs of flanges with a hammer and re-tightening the bolts will ensure maximum gripping power.

To True the Coupling

Chalk the pulley face of the coupling at the part which runs high, tighten the bolts nearest the chalk mark. and see that the space between the two flanges is uniform.

To remove the couplings take the nuts off bolts and insert a cold chisel between the rims, thus forcing the shells apart. Rap shells lightly and they will slide down the sleeves.

Dimension of Compression Couplings

	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Diameter of shaft .	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4
Outside diameter .	6½	7	8	8½	9½	10	10½	11	12	13
Overall length . .	5½	6	7½	8	9	9	10	10	11	12

Claw Couplings (Fig. 9)

When it is desired to connect or disconnect two lengths of shafting and both can be brought to rest for this purpose, claw-type couplings can be used.

If desired, this design of coupling can be constructed so that power is transmitted in one direction only (Fig. 10).

Flexible Couplings

These are an indispensable feature of modern power transmission, their primary object being to protect driving and driven machinery from

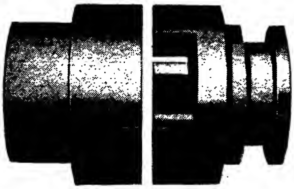


Fig. 9

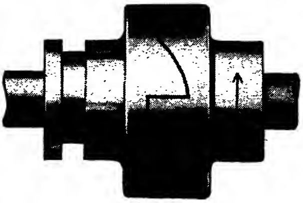


Fig. 10

Fig. 9.—CLAW COUPLING

Fig. 10.—CLAW COUPLING CONSTRUCTED SO THAT POWER IS TRANSMITTED IN ONE DIRECTION ONLY

Diameter of Bore. Inches.	*Overall Diameter. Inches.*	*Overall Length when in Clutch. Inches.*	*Maximum h.p. at 100 r.p.m. under favourable working conditions.*
1½ & 1¾	4½	5⅝	4
2 & 2¼	6	7½	8
2½ & 2¾	7½	9¼	15
3 & 3¼	9	11	25
3½	10½	12¾	50
4	12	14½	75
4½	13½	16¼	100
5	15	18	150
6	18	21½	250

detrimental effects, which may arise from misalignment of shafts, vibration, sudden shock loads, end float, or shaft expansion.

It will be easier to select the correct design of flexible coupling for any installation if it is clearly understood why the coupling is being installed.

It would be quite impossible to illustrate all the designs of flexible couplings at present on the market. In some designs the object is:

(1) To provide easily renewable driving surfaces, whilst in others it is
(2) To provide lubricated surfaces so as to minimise wear.
(3) To provide "resilience."

Belting, rubber, and ropes have been employed in all manner of ways to connect the driving and driven portions of couplings under class (1).

Couplings under classes (2) and (3) are *usually* (but not always) of all-metal construction.

It is usual for manufacturers to publish a list giving horse-powers which flexible couplings will transmit under favourable conditions, together with factors to modify this rating for arduous drives.

Selection

To determine the size of coupling required for a particular drive:

First find the normal horse-power at 100 revolutions per minute, using the following equation:

$$\text{Normal horse-power at 100 r.p.m.} = \frac{\text{h.p.} \times 100}{\text{r.p.m.}}$$

Having obtained the normal horse-power, multiply it by a factor from the list below to obtain the listed horse-power of coupling which will be suitable for the duty required.

Type of Driven Machine	*Type of Prime Mover*	
	Electric Motor	*Four-cylinder Gas- or Oil-engine*
	Factor	*Factor*
Generators, Centrifugal Pumps, etc.	1	2
Machine Tools, Workshop Shafting, etc.	1·5	2·5
Heavy Fan Drives, Winders, etc.	2	3
Compressors, Three-throw Pumps	2·5	4
Crushers, Dredgers	3	5

Equivalent Shaft

In many drives it is difficult to estimate what power or torque may have to be transmitted under every possible circumstance. In such cases the only safe plan is to take the strength of the smaller of the two shafts (based on a stress of 8,000 lb. per square inch) and use this as a basis for selection. Many manufacturers state the size of shaft which they consider is of equal strength to the listed coupling size.

The Crown-pin Type Coupling

This type, shown in Fig. 11, was made popular by Dr. De Laval, of steam-turbine fame, and is probably the most extensively used of all types of flexible couplings, being comparatively cheap and suitable for driving all classes of machinery where there is little prospect of excessive misalignment of shafts, as is the case when driver and driven units are mounted on a common bedplate. They will drive in either direction, allow for slight misalignment, expansion, and end float, have low maintenance cost, and can quickly be disconnected by removing the driving pins.

Construction

Two cast-iron or cast-steel flanged discs each having a number of steel studs fitted with rubber or leather buffers arranged to engage in alternate holes in the opposite half.

For capacities of this type of coupling, see Table III.

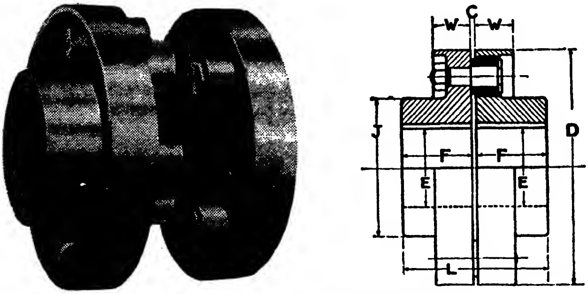


Fig. 11.—Crown-pin type coupling

Belt-type Flexible Couplings

These have been used for all general industrial drives. They take up slight

TABLE I.—TRANSMISSION SHAFTING FOR TRANSMITTING POWER ONLY

Approximate Limit of Distance Apart of Centres of Bearings for Shafts which do not *carry any Pulley, Gears, etc.*

Dia. of Shaft in.	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5	5½	6
Revs. per Minute	*Distance between Bearing Centres in Feet*													
100	9	10	11	11½	12	12½	13	13½	14	14½	15	16	17	18
150	8½	9½	10¾	11¼	11¾	12	12½	13	13½	14	14½	15½	16½	17½
200	8	9	10½	10¾	11¼	11½	12	12½	13	13½	13¾	14¾	15¾	16¾
250	7½	8½	10	10¼	10¾	11	11½	12	12¼	12¾	13¼	14	15	16
300	7	8	9½	9¾	10	10¼	10¾	11	11½	12	12½	13¼	14	15
350	6¾	7¾	8¾	9	9¼	9½	10	10½	10¾	11	11½	12¼	13	13¾
400	6¼	7¼	8	8¼	8½	9	9¼	9½	10	10¼	10½	11¼	12	12½

TABLE II.—LINE SHAFTING CARRYING PULLEYS, ETC.

Approximate Limit of Distance Apart of Centres of Bearings for Shafts which do not carry more than an Average Number of Pulleys transmitting Normal Powers

Dia. of Shaft in.	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5	5½	6
Revs. per Minute	*Distance between Bearing Centres in Feet*													
100	7¼	7½	8	8½	9	9½	10	10½	11	11½	12	13	14	15
150	6¾	7¼	7¾	8¼	8¾	9¼	9¾	10¼	10¾	11¼	11¾	12¾	13¾	14½
200	6½	7	7½	8	8½	9	9½	10	10½	11	11½	12¼	13¼	14
250	6	6¾	7¼	7¾	8¼	8¾	9¼	9¾	10	10½	11	11¾	12½	13¼
300	5¾	6½	7	7½	8	8½	8¾	9¼	9½	10	10½	11	11¾	12½
350	5¼	6	6¾	7¼	7¾	8	8½	8¾	9	9½	9¾	10¼	11	11½
400	5	5¾	6½	7	7½	7¾	8	8¼	8½	8¾	9	9½	10	10½

misalignment, angularity, end movement, etc. They are used particularly to transmit medium powers at low speeds, i.e. gas and oil-engine drives. The coupling illustrated in Fig. 12 is suitable for driving in one direction only.

Construction

The coupling consists of inner and outer halves of cast iron or cast steel connected by a tough flexible belt, the number and width of loops being determined by size and power requirements.

Special Note.—To ensure the utmost efficiency and long life the belt must be specially selected to suit the atmosphere in which it must work.

Method of Replacing Belt

Before commencing to thread the belt into its position, the outer and inner halves of the coupling should be centralised by means of a sufficient number of distance pieces " P " (Fig. 13). Draw the belt as tightly as possible around all the webs or lugs. By making use of wedges, as " W," the belt can be held secure whilst each loop is made taut. One of the outer

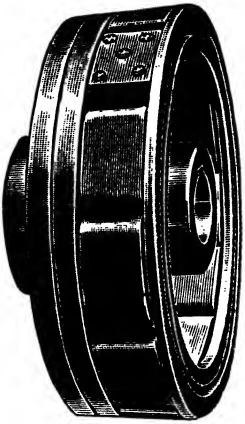
TABLE III.—DIMENSIONS AND HORSE-POWER TABLES FOR CROWN-TYPE COUPLINGS

Size of Coupling in Inches	*Max.h.p. at 100 r.p.m.*	*Equiv. Shaft. Diam. in Inches*	*Max. Safe Speed in Cast Iron*	*Max. Bore*	*Dimensions in Inches*					
					C	*D*	*F*	*J*	*L*	*W*
4½	1	¾	4479	1⅜	3/32	4½	1½	2 7/16	3 3/32	⅞
5½	2·7	1	3665	1¾	3/32	5½	1¾	2⅞	3 19/32	1⅛
6¼	4	1¼	3225	2	⅛	6¼	2	3¼	4⅛	1⅜
7½	8	1½	2690	2½	⅛	7½	2⅜	4 1/16	4⅞	1⅜
9	15	1⅞	2240	3	3/16	9	2¾	4¾	5 11/16	1¾
10½	22	2⅛	1920	3½	3/16	10½	3¼	6⅛	6 11/16	1¾
12	38	2½	1680	3¾	¼	12½	3¾	6½	7¾	2 1/16
13½	55	2⅞	1495	4½	¼	13½	4¼	8	8¾	2 1/16
15	78	3¼	1345	4¾	¼	15	4¾	8¾	9¾	2 7/16
16½	102	3⅜	1220	5½	¼	16½	5¼	10	10¾	2 7/16
18	135	3¾	1120	5¾	5/16	18	5¾	10½	11 13/16	2⅞
21	221	4½	960	7½	5/16	21	6¾	13¾	13 13/16	2⅞
24	340	5¼	840	8½	11/32	24	7¾	15¾	15 27/32	3 5/16
27	430	5⅝	745	10	11/32	27	8½	18¾	17 11/32	3 5/16
30	650	6½	670	11	⅜	30	9	20¼	18⅜	3⅞
36	980	7½	560	14½	⅜	36	10	26	20⅜	3⅞
42	1800	9¼	480	17	½	42	11	31	22½	4⅞
48	2500	10¼	420	20	½	48	12	37	24½	4⅞

series of webs or lugs is purposely made lower than the others by the thickness of the belt ; this arrangement permits of a uniform peripheral surface of the jointing with the remaining portion of the belt. When inserting belt, it should be started and finished on this low web or lug "L" (*see* Fig. 13).

All-metal-type Couplings (Resilient Design)

Fig. 14 shows a patented resilient-spring type coupling which is proving increasingly popular, and is applied universally to machinery and shafting drives up to the largest powers. It has adjustable flexibility and takes care of misalignment of shafts, strains due to yielding of supports, uneven wear in bearings, vibration, shock loads, etc., it is self-



Size of Coupling	*Max. H.P. at 100 R.P.M.*	*Equivalent Shaft*	*Bore*		*Dimensions in Inches*							*Max. Safe Speed in Cast Iron*
			Min.	*Max.*	*A*	*B*	*C*	*D*	*G*	*J*	*L*	
6½	2	1	¾	1⅜ & 1¾	6½	2 3/16	5/16	2 3/16	3½	4 11/16	3¼	2800
8	3·5	1⅛	1	1⅞ & 2	8	2½	5/16	2½	4	5 5/16	3⅝	2250
9½	5	1¼	1¼	2⅜	9½	2 13/16	⅜	2 13/16	4¾	6	4 1/16	1850
11	6·75	1⅜	1½	2¾	11	3⅛	⅜	3⅛	5½	6⅝	4½	1600
12½	7·75	1½	1¾	3¼	12½	3⅛	⅜	3⅛	6¼	6⅝	4½	1400
14	10·5	1¾	1¾	3¾	14	4	⅜	4	7¼	8⅜	5⅝	1200
15½	12	1¾	2	4½	15½	3⅞	½	3⅞	8¼	8¼	5⅝	1100
17	37	2½	2	5	17	4⅝	½	4⅝	9	9¾	6⅜	1000
18½	43	2⅝	2	5	18½	4⅞	½	4⅞	10	10¼	6⅝	900
20	53	2¾	2½	5½	20	5⅜	⅝	5⅜	10¾	11⅜	7½	850
21¾	58	3	2½	5½	21¾	5⅜	⅝	5⅜	12½	11⅜	7½	800
23	87	3¼	3	6	23	5⅞	⅝	5⅞	12½	12⅜	8	750
24½	105	3½	3	6	24½	5⅞	⅝	5⅞	14	12⅜	8	700
25½	125	3¾	3	7	25½	6½	¾	6½	14	13¾	8⅞	650
27	136	4	3½	7	27	6½	¾	6½	17	13¾	8⅞	625
28½	185	4¼	3½	8	28½	7 3/16	13/16	7 3/16	17	15 3/16	9⅝	580
30	195	4¼	4	8	30	7¼	¾	7¼	17	15¼	9⅝	540

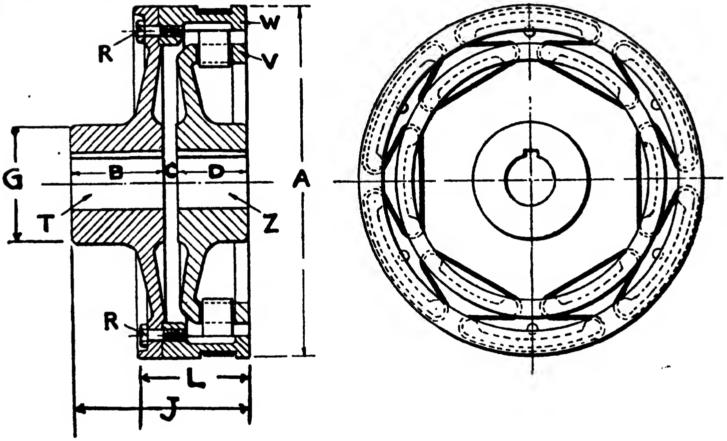


Fig. 12.—Belt-type flexible coupling

lubricating, and will run at high or low speeds in either direction of rotation.

Independent renewable connecting springs afford a means of positively altering the elastic strength of the coupling should there be resonance in the shaft system.

Construction

It is of all-metal construction, consisting of two flanged discs having accurately pitched grooves cut in the outer peripheries, one disc with parallel, and the other with inwardly flared grooves of special form connected by special chrome-vanadium springs fitting in corresponding

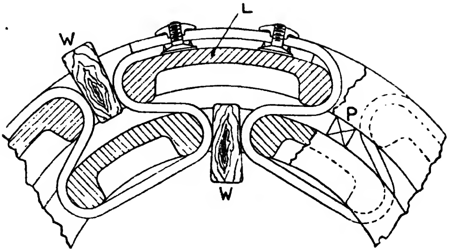


Fig. 13.—Diagram showing method of inserting belt in a coupling

grooves in the two discs. These form the resilient transmission members. Semi-steel, oil-tight, dust-proof covers totally enclose the springs. Under normal loads the springs have a long flexible span; as the load increases, greater support is afforded to the springs by the walls of the flared grooves, thus shortening the span and reducing to a minimum the ultimate stresses in the springs. The use of independent springs enables flexibility to be adjusted by removing opposite springs when required to increase flexibility.

All-metal-type Couplings (Gear Type)

Another design of all-metal coupling which is being extensively used for heavy drives such as rolling mills, cement mills, etc., is the *internal gear type* (Fig. 15).

These are designed to compensate for shaft misalignment, with the particular advantage of allowing end movement without transmitting longitudinal thrust; they are self-lubricating, the method employed ensuring that all engaging surfaces are submerged in oil.

The standard type consists of two cast-steel hubs with teeth cut in their outer peripheries, the hubs being keyed respectively to the driving and driven shafts, thus forming two spur rings which mesh with internal gears cut in the inner faces of annular oil-tight dust-proof covers, bolted together to form a totally enclosed oil reservoir.

By removing bolts and moving covers endwise either shaft can be lifted clear.

Horse Powers and Approximate Dimensions

Size of Coupling	*Max. Bore*	*Equivalent Shaft*	*Safe Speed*	*Max. H.P. at 100 r.p.m.*	*Size of Coupling*	*Max. Bore*	*Equivalent Shaft*	*Safe Speed*	*Max. H.P. at 100 r.p.m.*
4¼	1¼	⅞	4750	1·5	20	6	4¾	950	250
5¼	1¾	1⅛	3800	3·0	21½	7	5	850	274
6¼	2	1½	3100	8·0	23	8	5¼	800	350
7¼	2	1¾	2700	10·0	25	8½	5¾	750	430
8	2¼	1¾	2350	12·0	27	9	6	700	510
9	2½	2	2100	16·0	30	10	7	600	850
10	2¾	2¼	1900	23·0	33	11	7½	550	1000
11	3¼	2½	1700	42·0	36	12	8½	500	1500
12½	3½	3	1500	63·0	42	15	9½	450	2250
14	4¼	3¼	1350	86·0	48	17	11	400	3000
15½	5	3½	1200	97·0	54	19	12	350	4000
17	5¼	3¾	1100	130·0	60	21	13	300	5000
18½	5½	4	1000	150·0					

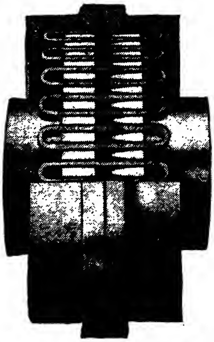


Fig. 14.—Improved multiflex coupling (Crofts, Ltd.)

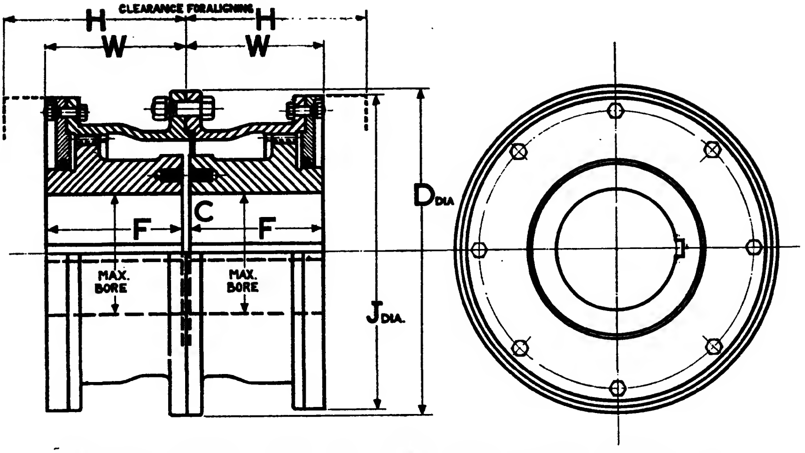


Fig. 15.—Internal gear-type flexible couplings (Crofts, Ltd.)

Powers Transmitted. Dimensions

Coupling Ref. No.	*Max. Bore*	*H.P. at 100 r.p.m.*	*Max. Speed in r.p.m.*	*Dimensions in Inches*						*Oil Capacity in Gals.*	*Net Weight in Lbs.*	
			Steel	*C*	*D*	*F*	*H*	*J*	*W*		*C.I.*	*Steel*
6	6½	500	2400	5/16	16	6¼	7¼	15¼	6 13/32	¼	325	350
7	7½	800	2200	⅜	18¼	7	8	17½	7 3/16	½	510	550
8	8½	1200	2000	⅜	21	8	9⅛	20	8 3/16	1	745	800
9	9½	1600	1900	½	23	9	10¼	22	9¼	1½	1030	1100
10	11	2200	1800	½	26	10	11½	24¾	10¼	2	1300	1400
11	12	3000	1600	½	28	11	12½	26¾	11¼	2½	1670	1800
12	13	3800	1400	½	30¼	12	13½	29	12¼	3	2230	2400
14	15	6000	1200	¾	35½	14	15¾	33¾	14⅜	4	3350	3600
16	17½	8800	1000	¾	39	16	18	37¼	16⅜	5	4650	5000
18	20	10000	700	1	44¼	18	20½	42¼	18½	6½	6320	6800
20	22	15000	500	1	48½	19	22	46½	19½	8	8375	9000
22	24	20000	400	1	51½	19	22	49½	19½	10	11200	12000
24	26	30000	300	1	54	19	22	51½	19½	14	16000	17000

Erection Instructions for all Types of Couplings

Correct alignment of couplings is essential to ensure satisfactory driving; it is important, therefore, to take every care that couplings are fitted correctly and that shafts are perfectly aligned before starting up. It is advisable also to check alignment at regular intervals and to correct misalignment due to settlement of foundations, wear in bearings, or any other cause.

Flexible couplings are not intended to overcome serious or permanent misalignment of shafts which must be rectified at the earliest moment.

Fitting

Both halves of couplings should be securely keyed to their respective shafts, using parallel or taper keys with sunk keyways in the shafts. The arrangement shown in Fig. 16 can be used for forcing couplings on to shaft ends, using a tube with a larger inside diameter than the shaft end. After keying up, shafts should be checked for alignment, and in the case of rigid couplings, it is recommended that they should be faced up on the shafts after fitting keys.

Alignment

Flexible couplings should be perfectly aligned ; this can be checked by allowing each half to rotate independently and inserting a feeler or suitable-thickness gauge at points shown on diagram (Fig. 17) at *A*, *B*, *C*, *D*. Rotate one half coupling from position *A* to *B*, *B* to *C*, and *C* to *D*, testing in each position. By this method, alignment of both shafts can be checked for accuracy.

Level

To check level of shafts, rotate each half independently, using straight-edge as illustrated in Fig. 18 at each position *A*, *B*, *C*, *D*. This will ensure correct level being obtained.

Withdrawal of Couplings

After removal of connecting mediums, springs, bolts, etc., the respective coupling halves may be withdrawn from the shafts by either of the following methods :

(*a*) If coupling half is fitted with tapped holes on face, insert a distance-piece of smaller diameter

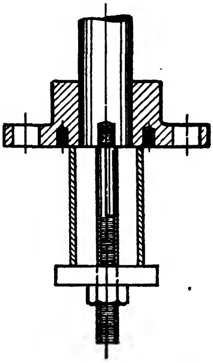


Fig. 16.—DIAGRAM SHOWING METHOD OF FORCING COUPLINGS ON SHAFTS

Using a tube with a larger inside diameter than the shaft end.

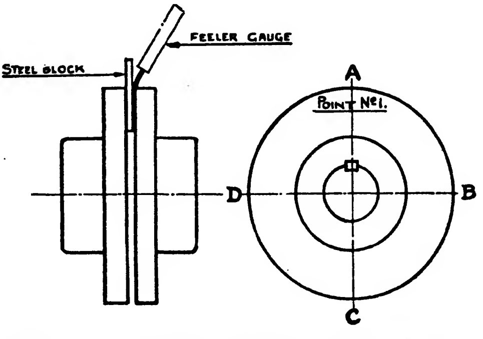


Fig. 17.—DIAGRAM SHOWING ALIGNMENT OF FLEXIBLE COUPLINGS

This method is suitable for coupling having machined flanges.

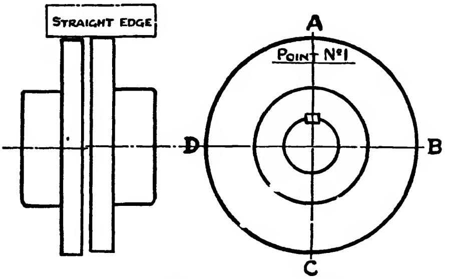


Fig. 18.—DIAGRAM SHOWING METHOD OF LEVELLING COUPLINGS

This method is suitable for couplings having machined flanges.

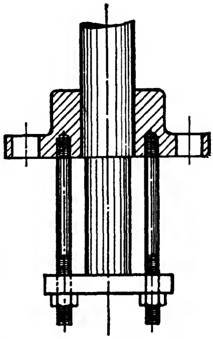


Fig. 19.—DIAGRAM SHOWING METHOD OF WITHDRAWING COUPLINGS FROM SHAFT

This is a suitable method where the coupling flanges have tapped holes.

than the shaft and screw up against the cross-plate as shown in Fig. 19.

(*b*) An alternative method is driving the coupling off from the rear, taking care to insert a wood block behind the coupling to avoid damage.

Where couplings are rusted on, they can generally be removed by soaking in vinegar, paraffin, methylated spirits, or warming round the boss with a blow-lamp.

Shaft Bearings

These form one of the most considerable factors in power transmission, and upon their suitability or special fitness for the work on which they are employed depend far-reaching economic results.

Bearings operating under ordinary working conditions warm up until the heat radiated equals the heat generated and will remain at this temperature provided there is no change made in lubrication, load, or speed.

The rise in temperature over that of the surrounding atmosphere varies from 10 to 100° F., but is generally about 30° F.

Excessive pressures, such as will squeeze the oil out of the bearing, must be avoided, as the absence of lubrication will cause abrasion of the materials in contact and an immediate rise in temperature, which will cause the internal parts of the bearing to expand more rapidly than the external parts, causing the bearing to grip the journal in such a manner as to seize.

The pressure which any bearing will withstand without seizing depends largely on the temperature, velocity of rubbing, and quality of the lubricant.

A very light oil is liable to be squeezed out of the bearing with a pressure as low as 50 lb. per square inch, whilst, on the other hand, there are oils now sold which will stand a pressure of 500 lb. per square inch or more ; the advantage of selecting suitable oil is therefore apparent.

Proper lubrication of bearing surfaces involves careful consideration of materials comprising the journal and its bearings. The bearing surfaces in general use come mainly under the following heads : Cast iron, steel, various alloys of bronze, and babbitt. Cast-iron bearing surfaces should only be used for low speeds and light pressures. All normally loaded bearings should have continuous lubrication, and the simplest and cheapest method of obtaining a copious and regular supply of oil is by loose oil rings, as Fig. 4.

Heavily loaded or high-speed bearings are sometimes fitted with pumps to force oil on to the friction surfaces and to keep a large volume of oil in circulation, thus preventing overheating by the additional radiating area of the pipes and oil reservoir.

BALL AND ROLLER BEARINGS

These are now being extensively installed, and when properly mounted can be used with confidence.

It is important to remember that provision must be made for expansion of the shaft, and that although lubrication is only required at intervals of about three months, it is an important matter, but it should be understood it is mainly intended to prevent rusting of the polished working surfaces.

It is absolutely essential to keep the working surfaces clean. If grit and dirt find their way into the bearing, they are sure to reduce its life ; it is therefore usual to adopt some method of sealing the bearing to prevent the ingress of foreign matter, liquids, etc. A series of small annular grooves turned in the bore of the cover will, when full of grease, form an effective dust-seal, or felt pads may be fitted, as in Fig. 20.

The outstanding feature of ball and roller bearings is their low starting friction, which is practically the same as running friction ; therefore, if a shaft must frequently be started from rest, it is worth while considering the adoption of ball or roller bearings to support it, as they will no doubt effect a considerable saving in power and lubricant. The higher the speed, the greater will be the saving.

As it is almost impossible to maintain adequate lubrication of plain bushes in loose pulleys, ball bearings show up to advantage in this class of installation, and when correctly mounted can be considered as a certain cure for the troubles which are so common with loose-pulley bushings.

A loose pulley mounted on ball and roller bearings is shown in Fig. 21.

The Erection of Line Shafting

Although the self-aligning properties of ball bearings compensate for certain faults in the adjustment, the bearings do not give the good results one is justified in expecting, if they are carelessly mounted. We are

Fig. 19A (*Left*).—**Levelling a wall bracket**

First operation.

indebted to the Skefko Ball Bearing Company Ltd., for the following notes on the erection of SKF power transmission equipment.

The first operation is to place the hangers, wall brackets, etc., in their exact positions with the aid of a spirit level and a taut line.

Fig. 19B (*Right*).—**Levelling a wall bracket**

Second operation.

By means of a steel set-square, the positions of the wall brackets are adjusted and the hangers suspended exactly at right angles to the axis of the shaft.

The positions of the brackets and hangers are checked horizontally and vertically by means of plumb-lines.

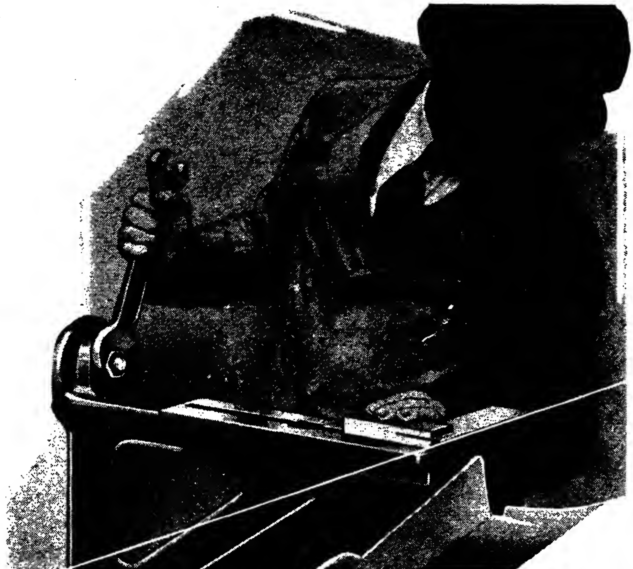


Fig. 19C (*Left*).—ADJUSTING POSITION OF A WALL BRACKET BY MEANS OF A STEEL SET SQUARE AND A TAUT LINE

When the housings have been cleaned, the felt washers are saturated with oil, divided and placed in the grooves in the housings, care being taken to ensure that the divided ends of the felt washers do not protrude between the faces of the cap and lower section.

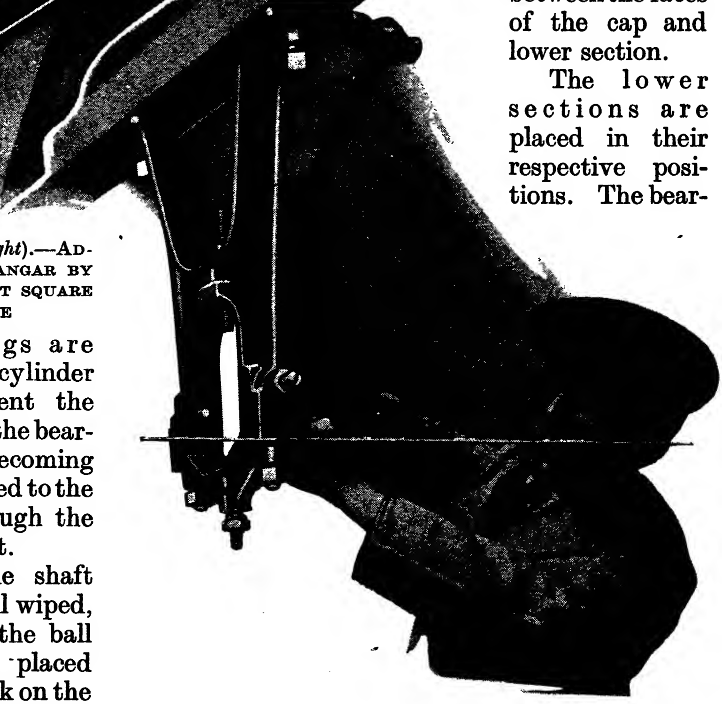


Fig. 19D (*Right*).—ADJUSTING A HANGAR BY MEANS OF SET SQUARE AND TAUT LINE

The lower sections are placed in their respective positions. The bearing seatings are painted with cylinder oil to prevent the outer ring of the bearing from becoming firmly attached to the housing through the action of rust.

When the shaft has been well wiped, the parts of the ball bearings are placed upon it. Look on the end of the shaft from

Fig. 19E.—Placing the felt washers in the grooves in the housings

which the parts have to be threaded on, and if the shaft is to turn clockwise, first put on the nut, tab washer, bearing and, lastly, the adapter sleeve; if the shaft is to turn anti-clockwise, first thread on the sleeve, bearing, tab washer and, lastly, the nut. This ensures that the nuts tighten up in the opposite direction to rotation of the shaft.

Notes.—See that the concave side of the tab washer faces the nut.

Do not drive the sleeves too tightly into the bearings. It will be found easier to move the sleeve along the shaft if a screwdriver is forced into the slit of the sleeve.

When keyed couplings or pulleys are being fitted, do not rest the shaft on the bearings in such a manner that heavy blows can be transmitted to the bearings.

The shaft with bearings is placed in the lower sections of the housings and a spirit level is used to determine whether the shaft is perfectly horizontal. Tap lightly round the sides of the outer rings to ensure that they seat correctly; do not force them into place by means of the cap. Caps are not interchangeable.

The bearings are secured in their respective positions by tightening the sleeve nuts with a spanner, the located bearing being positioned so as definitely to locate the shaft endwise.

In the case of " free " bearings, care is taken to make sure that the bearing fits into its seating in such a manner that it can move sideways with complete freedom when the shaft expands or contracts.

For normal temperature changes no " free " bearing housing should

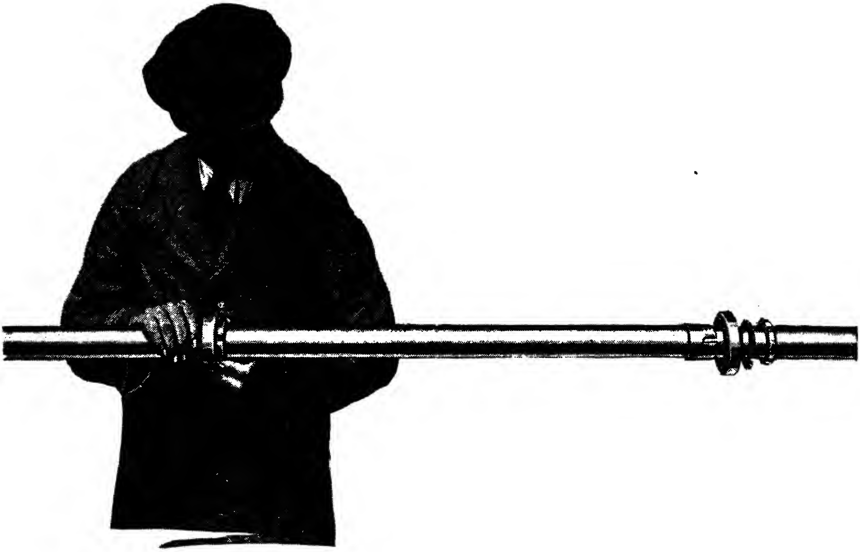


Fig. 19F.—Assembling the parts of the ball bearing

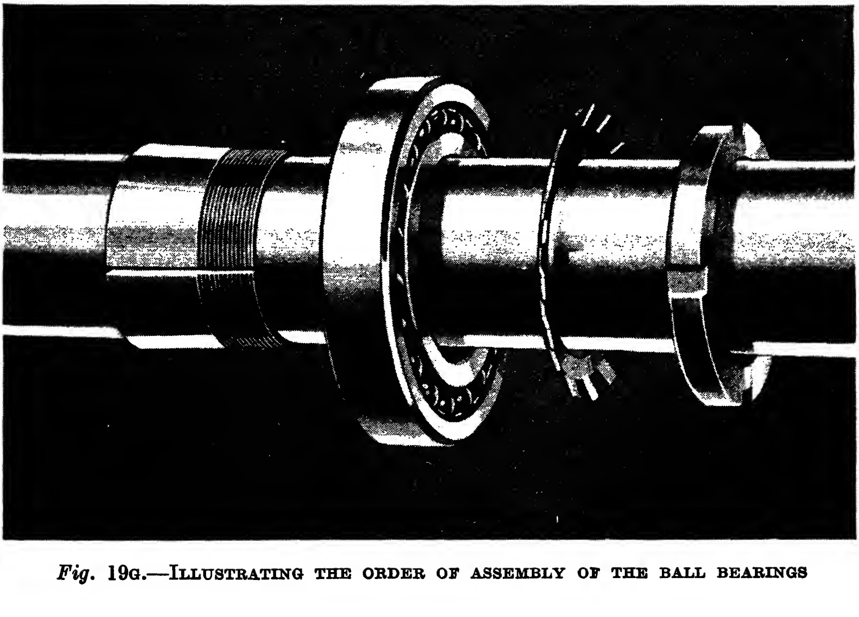


Fig. 19G.—Illustrating the order of assembly of the ball bearings

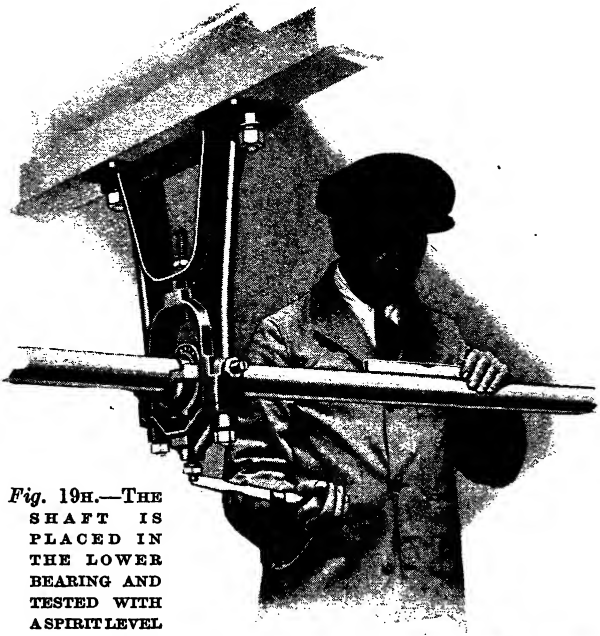


Fig. 19H.—THE SHAFT IS PLACED IN THE LOWER BEARING AND TESTED WITH A SPIRIT LEVEL

be further from the located bearing housing than 70 ft. unless special housings are used.

It is usually advisable, particularly where there is vibration, to check over the tightness of the sleeve nuts after the shafting has been running for a short period.

The sleeve nuts are locked by means of the safety tab washers, one of the outside tabs being bent over into the appropriate slot in the nut.

A located bearing registers the shaft endwise, this being effected by inserting a locating ring (or rings) in the housing as illustrated. Only one located bearing should be used on each rigidly coupled shaft; the position of this is usually important and it should be placed according to special instructions. A "free" bearing has no "locating ring" in the housing.

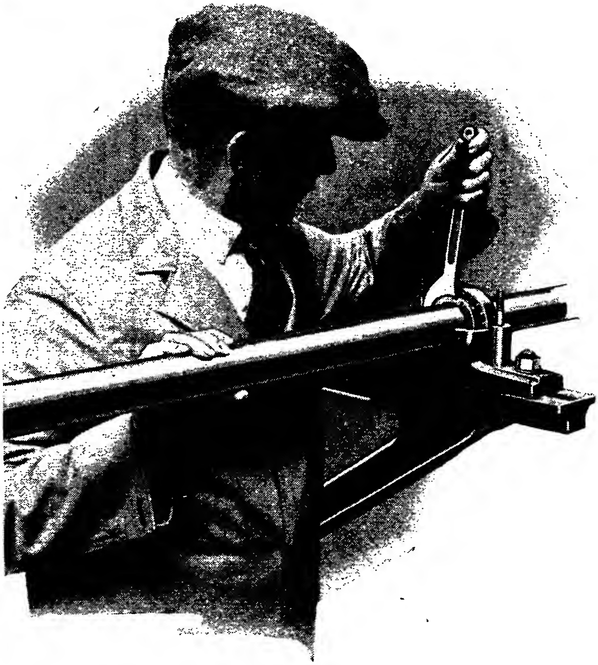


Fig. 19I.—THE BEARINGS ARE SECURED IN THEIR RESPECTIVE POSITIONS BY TIGHTENING THE SLEEVE NUTS

Fill about three-quarters of the available space in the complete housing with a recommended ball-bearing grease.

The bearing seatings in the housing caps are painted with cylinder oil. The caps are then placed in their respective positions. In the case of hangers, care should be taken to avoid tightening the adjusting screws to such an extent as to cause distortion of the bearing housing. This is of the utmost importance.

By means of a double plumb-line and a scale it is ascertained whether the shaft throughout its length lies in the same vertical plane. Readjustment with a spirit level is also necessary in most cases.

Finally, the shaft is carefully examined to make sure that it

Fig. 19J.—LOCKING THE SLEEVE NUTS BY MEANS OF THE SAFETY TAB WASHERS

Fig. 19K.—INSERTING A LOCATING RING

does not foul the housings at any point.

The above series of operations is illustrated in Figs 19A–19P.

General Notes on Mounting, etc.

It is absolutely essential to see that the shaft is perfectly round and true, and that the bore of the pulley or bearing housing is perfectly cylindrical. This point is of special importance in the case of split pulleys

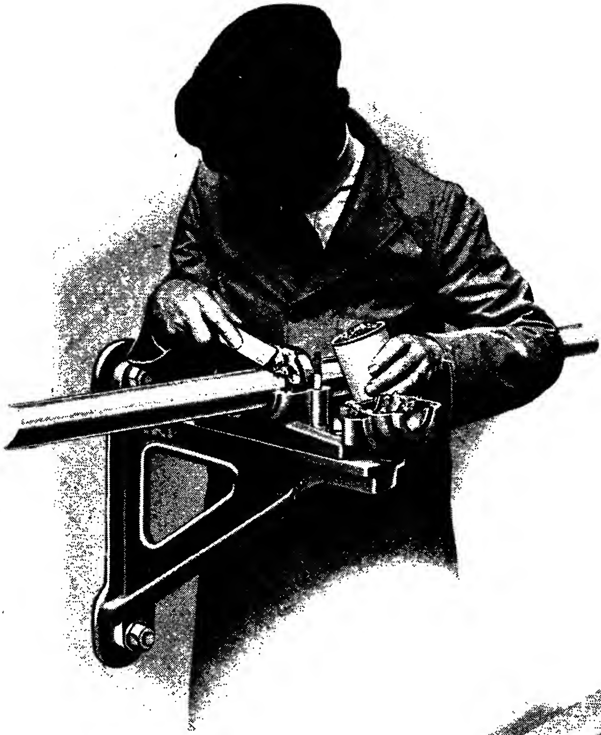


Fig. 19L (*Above*).—FILL ABOUT $\frac{3}{4}$ OF THE AVAILABLE SPACE IN THE COMPLETE HOUSING WITH BALL-BEARING GREASE

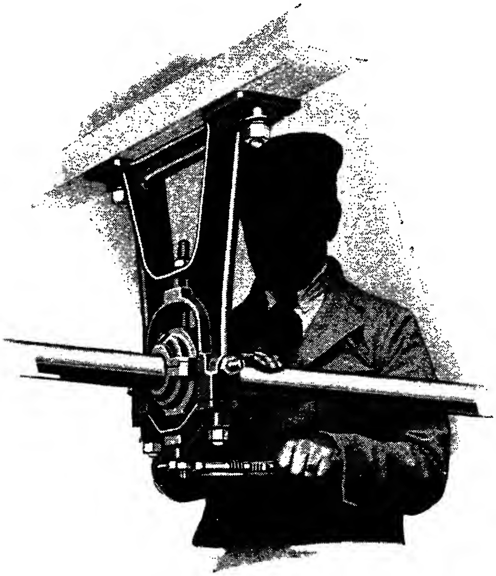


Fig. 19M (*Right*).—IN THE CASE OF HANGARS AVOID TIGHTENING THE ADJUSTING SCREW TO SUCH AN EXTENT AS TO CAUSE DISTORTION

Fig. 19N (*Right*).—TESTING WITH DOUBLE PLUMB-LINE AND SCALE

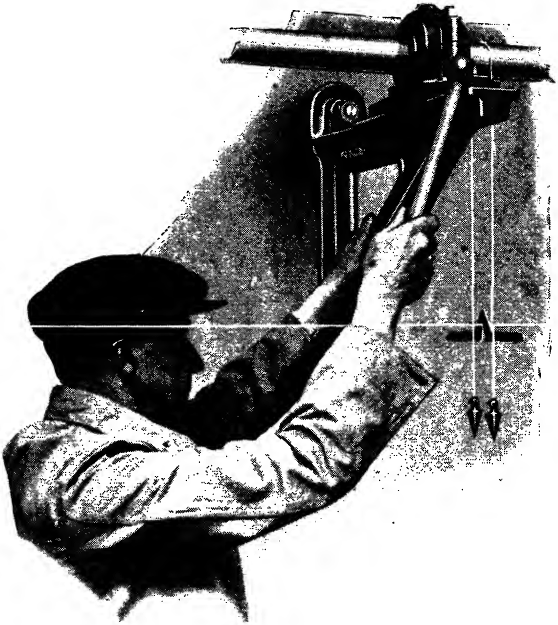
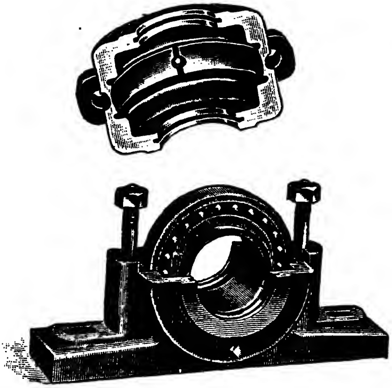


Fig. 19P (*Left*).—FINALLY EXAMINE SHAFT TO MAKE SURE IT DOES NOT FOUL HOUSINGS AT ANY POINT



Dia. of Shaft	*Height to Centre*	*Centres of Bolt Holes*	*Length of Base*	*Width of Base*	*Width of Housing*	*Dia. of Bolts*	*Thickness of Base*
in.	*in.*	*in.*	*in.*	*in.*	*in.*	*in.*	*in.*
¾	—	—	—	—	—	—	—
1	2	5½	7	2	3	½	¾
1¼	2⅛	6¼	8	2¼	3	½	¾
1½	2¼	6½	8¼	2⅜	3⅜	½	¾
1¾	2½	6⅞	8½	2½	3½	½	⅞
2	2¾	8¼	10½	2¾	3¾	⅝	1
2¼	3	8¾	11½	3	4 5/16	⅝	1 1/16
2½	3⅛	9⅛	11⅛	3	4½	¾	1⅛
2¾	3½	10¼	12⅝	3¼	4⅝	¾	1¼
3	3¾	10⅝	13	3½	4¾	¾	1⅜
3¼	4	11¼	14	3¾	5⅛	⅞	1⅜
3½	4¼	12⅜	15½	4¼	5½	⅞	1½
3¾	4½	13	16	4⅜	5¾	1	1⅝
4	5	13½	16½	4½	6	1	1¾

Fig. 20.—SPLIT PEDESTAL BALL-BEARING PLUMMER BLOCK

and bearings with loose caps, as the outer and inner races are easily distorted and the least deformation will nip the balls, with disastrous results.

For ordinary millwright work the inner race is best made with a conical bore, clamped to the shaft by means of a taper sleeve and nut; the sleeve has a number of longitudinal slots, so that when the nut is tightened up the sleeve grips the shaft as well as the bearing. An efficient means must be provided to prevent the nut coming loose.

Lubricant

Use only good quality grease of consistency rather greater than vaseline and which contains no solidifying matter; a simple way of replenishing the grease when required is by means of a Stauffer lubricator fitted to each housing and delivering the grease near to the bearing. Avoid using too much grease.

Fixings for Shaft Bearings

It is important to ensure that fixings which support shaft bearings are capable of maintaining the shaft in correct level and alignment, and designed to enable errors in alignment to be easily corrected.

Vertical Adjustment

For heavy drives it is usual to mount the bearings on soleplates, as in Fig. 22, and to obtain the correct height by inserting iron or hard-wood packings between the soleplate and underside of the bearings.

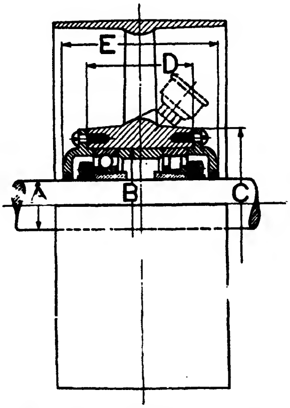


Fig. 21.—LOOSE PULLEY MOUNTED ON BALL AND ROLLER BEARINGS

Fig. 22.—Standard cast-iron sole plate

Suitable for Plummer Block .	. in.	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5	5½	6
Overall Length of Base .	. in.	16½	16½	19½	19½	23	23	26	26	27	30	34	36	39	42
Width of Base .	. in.	3⅝	3⅝	4½	4½	5½	5½	6½	6½	7	8	9	10	11	12
Distance from Base to Seating	. in.	1¼	1¼	1½	1½	1¾	1¾	2	2	2	2¼	2½	2¾	2¾	3¼
Centres of H.D. Bolts .	. in.	14	14	17	17	20	20	23	23	24	26½	30	32	34½	37 × 6

When adjustment for wear (or due to sinking of foundation) is necessary, fresh packings or steel " shims " can be inserted.

Horizontal Adjustment

It is usual to have cored slots in the soleplate and bearing to allow the latter to be fixed in the correct position after the soleplate has been securely bolted down to the foundations.

When the bearing has been finally bolted in place iron packings are placed between the ends of the bearing base and the " joggles " so that the holding-down bolts are relieved of any pull which may occur due to gears, belts, ropes, etc.

For exceptionally heavy drives, iron wedges are sometimes placed under the bearing, and set-screws fitted to locate same sideways (*see* Fig. 23).

Shafting which must be carried some distance from the floor and near to a wall can be supported by wall

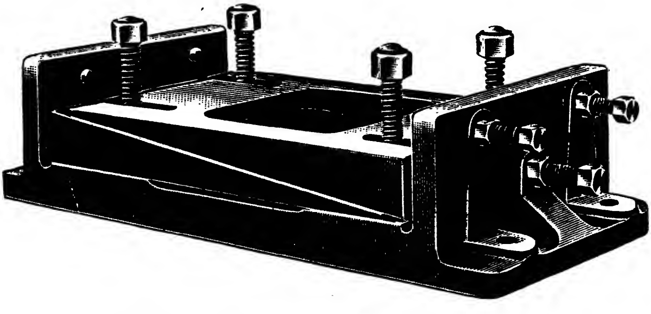


Fig. 23.—Wedge adjustment sole plate

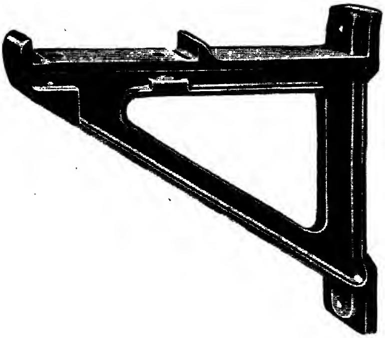


Fig. 24.—Wall bracket

brackets, as in Fig. 24, and packings inserted under the bearings as previously mentioned.

In cases where the wall is insufficiently strong to support the shafting special floor stands or pilasters can be used (*see* Figs. 25 and 26).

Overhead beams may be utilised to support sling hangers (Fig. 1), and it will be noted that it is not necessary to drill the beam. The writer considers that important building supports of wood or steel should never be drilled, for shaft brackets, etc., as this may weaken the building and is entirely unnecessary if the correct design of fixing is used.

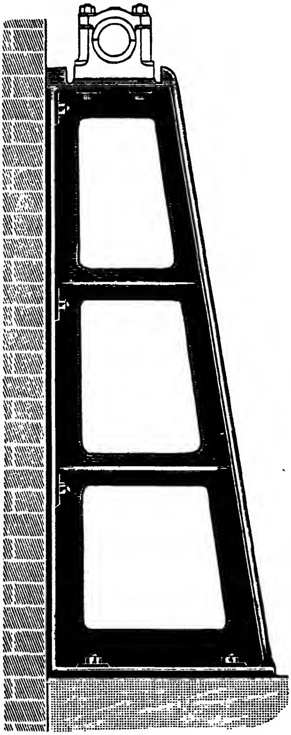


Fig. 25.—Floor stand

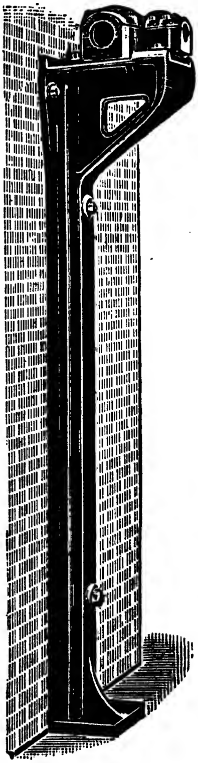
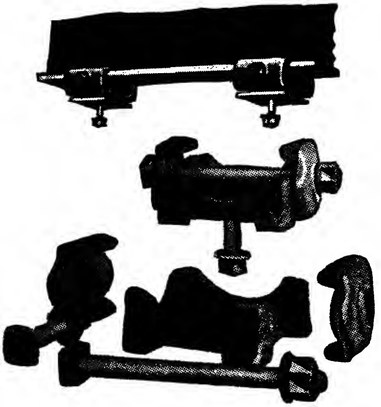


Fig. 26.—Pilaster



Width of Girder . in.	3 & 3½	4 & 4½	5
Distance from Base of Girder to Seating of Clamp . . . in.	2¼	2¼	2¼
Width of Girder . in.	6	7	8
Distance from Base of Girder to Seating of Clamp . . . in.	2¼	2¼	2¼

Fig. 27.—GIRDER CLAMP

Fig. 27 illustrates a neat and effective girder clamp which enables standard fixings to be used without any drilling of the girder.

In order to facilitate erection of shafting which is not called upon to sustain heavy pulls or gearing, it is permissible to use the Sovryn type of fixing (Fig. 28), which allows liberal adjustment in four directions and can be used as hanger, floor stand, or wall bracket.

Pulleys

Pulleys are made in various designs and of different materials, according to the speed at which they must run and the power to be transmitted.

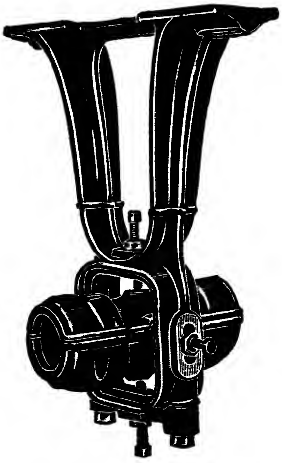


Fig. 28.—SOVRYN HANGER

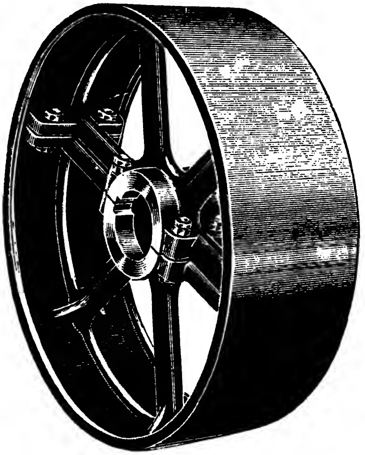


Fig. 29.—SINGLE-ARM SPLIT PULLEY

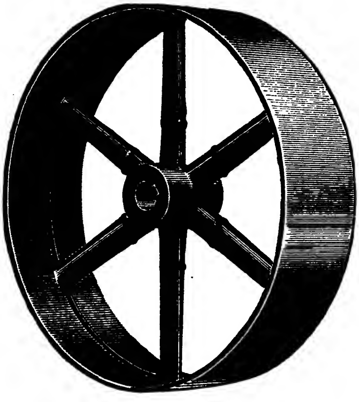


Fig. 30.—STANDARD CAST-IRON PULLEY

In some instances, generally owing to the speed of the driven machine, it is necessary to have comparatively large pulleys, although only a small power is to be transmitted, and for such cases wood or pressed-steel pulleys are useful, as they are light and cheap.

For medium and heavy drives it is usual to adopt cast-iron pulleys; ordinary designs are usually suitable for rim speeds up to 4,000 ft. per minute. For higher speeds (up to 5,500 ft. per minute) it is usual to make the pulley so that it can be accurately balanced. Fig. 29 shows a single-arm split pulley.

When several pulleys are placed on a shaft additional load is put on the bearing, and this is an important point in favour of wrought-iron pulleys, which are comparatively light and strong. The usual design of wrought-iron pulley has a cast-iron hub with wrought-iron rim ; pulleys up to 10 in. wide have one row of arms, wider pulleys may have two or three rows of arms, according to duty. For speeds over 4,000 ft. per minute and for arduous duties it is usual to adopt a form of treble-riveted construction, which gives more stiffness to the rim (*see* Figs. 30–31).

Small high-speed motor pulleys are usually made of cast iron and of plate-centre construction (*see* Fig. 33).

Pulleys should be wider than the belt, and flanged pulleys should be avoided wherever possible, as the flanges serve no useful purpose as far as actual driving is concerned, but frequently turn up the edges and ruin the belt.

Pulley rims should be slightly convexed, a pulley 12 in. wide should be increased in diameter at the centre not more than $\frac{1}{4}$ in. Fast and loose pulleys should have slightly crowned faces, but the pulley from which they receive their drive should be flat.

Rope pulleys of all kinds are usually made of cast iron, and it should be understood that the grooves must be accurately turned and the pulley carefully balanced if subsequent trouble is to be avoided.

Chapter II

PRACTICAL NOTES ON FITTING BELTS

IN this chapter we shall deal mainly with flat leather and balata belting. The important subject of "V"-rope drives is covered in the succeeding chapter.

To Calculate Length of Belt

To calculate the length of belt required for a certain drive, the following is a good rule : Add the diameters of the two pulleys together, divide the result by two, and multiply the quotient by 3·1416 ; then add this product to twice the distance between the centres of the shafts. The result is approximately the length required. A slight extra allowance must be made where pulleys have a large ratio.

Finding Length of Crossed Belt

To find the length of a crossed belt, mark off by lines the pulleys, set at their proper distances apart (either full size or accurately to scale), and measure the length of the side of the belt, supposing the belt to envelop one-half of the circumference only of each pulley, and add to this one-half the circumference of each pulley. If there is a great difference between the relative diameters of the pulleys and the distance apart of the shafts is unusually small, the length of the straight sides of the belt should be measured and the arcs of contact around the pulleys stepped off by compasses, the set of these not to be more than one-tenth the circumference of the pulley. In the case of both running horizontally, the upper half of the belt should be the driving side.

Finding Length with Tape Line

Many belt fitters of experience always ascertain the length of belt by passing a tape line over both pulleys, and after attaining this, where the belt happens to be fabric or balata, cut out ¾ in. for every 10 ft. of length in order to get the exact length of belt as it should be before it is fitted, e.g. for a drive the tape measurement of which is 100 ft., the belt is cut to 99 ft. 4½ in. long. Belt stretchers are used to pull the ends together over the 7½ in. gap so that they may be joined. This is only for drives with fixed centres.

For drives with adjustable centres, the measurement should be taken for the dead length of the belt before sliding the pulley forward, marking its position before doing so. As in the previous case, the belt should be cut ¾ in. short for each 10 ft. of tape measurement.

When the belt has been fitted and the driving pulley screwed back in its bed to the marked position, the belt will be stretched to the required tension.

The above method has been found through long experience to be very successful for all ordinary drives; that is to say, where there is no freak factor requiring an excessively tight or slack belt.

Belts with speeds over 4,000 ft. per minute should be installed to the exact dead-length measurement.

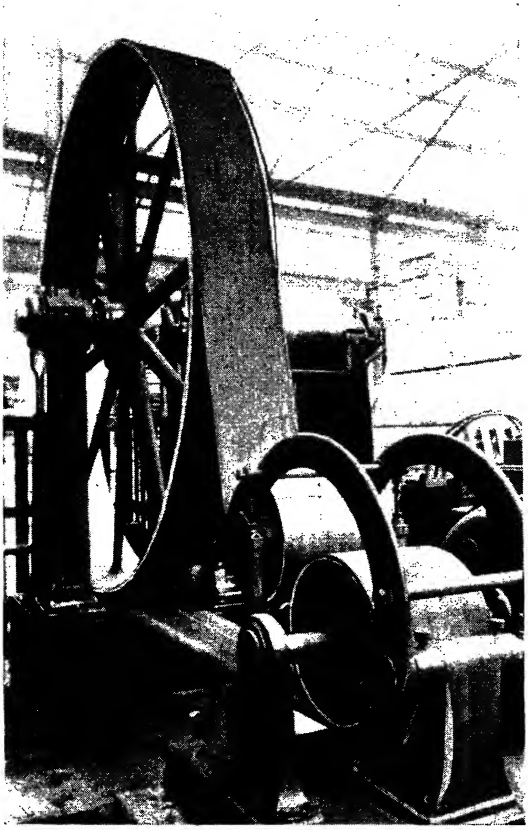


Fig. 1.—LARGE-RATIO DRIVE AT SHORT CENTRES WITH JOCKEY PULLEY

The jockey pulley tightens the belt round the pulley and increases the arc of contact. The drive is capable of 150 h.p. and a belt speed of 5,100 ft. per minute. The balata 6-ply belt is 16 in. wide.

How to Make the Belt Joint

The joint is the weakest part of a belt, and therefore requires the greatest consideration. There are three methods which are universally used, namely, lacings (principally used on leather belts), metal fasteners of various kinds, and endless, cemented, or solutioned joints.

Excellent results can be obtained from both metal and lacing joints, but the author has a preference for the cemented joint, which, when properly made, constitutes what is really an endless belt, and eliminates the many disadvantages to which a badly made metal or lacing joint is subject.

Endless Joint—Amount of Lap

The operation of making an endless joint is, however, a job for a skilled man. It at first appears difficult, but is simple once the method is learnt.

The first thing is to decide on the length of the lap required, and the following table gives the sizes for balata belting.

From 1 in. to 2 in. wide up to 10 ft., 6-in. lap.

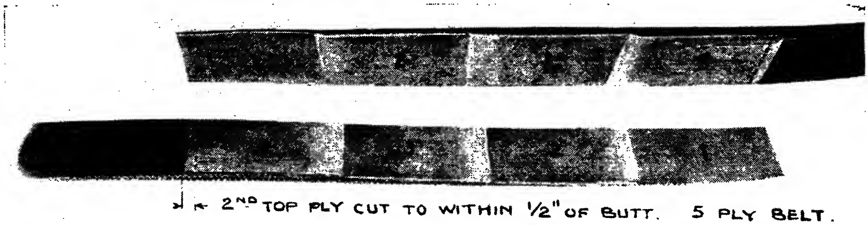


Fig. 2.—SPLICE FOR 5-PLY BALATA BELT

With lap of 12 in. Showing how the two ends are stepped for cementing together to make an endless belt. Note how the second top ply is cut back to within ½-in. of the butt, the top layer being cut to within 3 in. of the butt.

Over 10 ft. lengths up to 12 in. wide, 12-in. lap.
Over 12 in. wide, length for breadth, i.e. 14 in. wide, 14-in. lap, etc.

Splicing Balata Belt

To splice a 5-ply balata belt with a 12-in. lap, first pare round edge back to 12 in. The red covering matter is then scraped from the belt ends back to the mark or butt of splice. The belt end is then heated gradually at a stove or by means of heated irons to soften it for opening up. The top ply is removed to within 3 in. of the butt, afterwards the second ply is removed to within ½ in. of the butt (in every case of multiple plies the second ply is removed in this manner). The third ply is removed to within 6 in. of the butt, and the fourth ply to within 9 in. (Fig. 2). The edges of the steps to be pared down to avoid any abruptness in the finished joint. Roughen the surface of the steps to allow solution to penetrate the fabric.

Solution the Ends

Prepare both ends of belt as above, and they are now ready for the solution, which can be obtained from the belt manufacturers. Brush well into the fibre with a stiff short-haired brush, give two coats, the first light and the other heavy, and allow to stand for thirty minutes.

This is to allow the solvent to evaporate, and must be adhered to. The ends are then heated for a second time, and the ply marked 4 is stuck down over the ½-in. piece where the ply was removed.

The joint is now ready for the final heating, which must be gradual and not above 150° F. in case of solution catching fire. The ends are pressed down carefully and squarely and placed in a press (Fig. 5), which must be first wetted to prevent belt from sticking. The splice should remain in

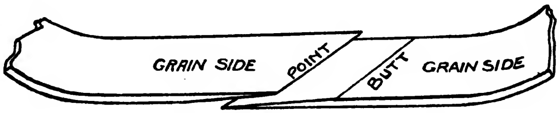


Fig. 3.—ENDLESS LAP FOR SINGLE LEATHER BELTS

Belt is tapered at each end. The ends are then joined with glue or cement.

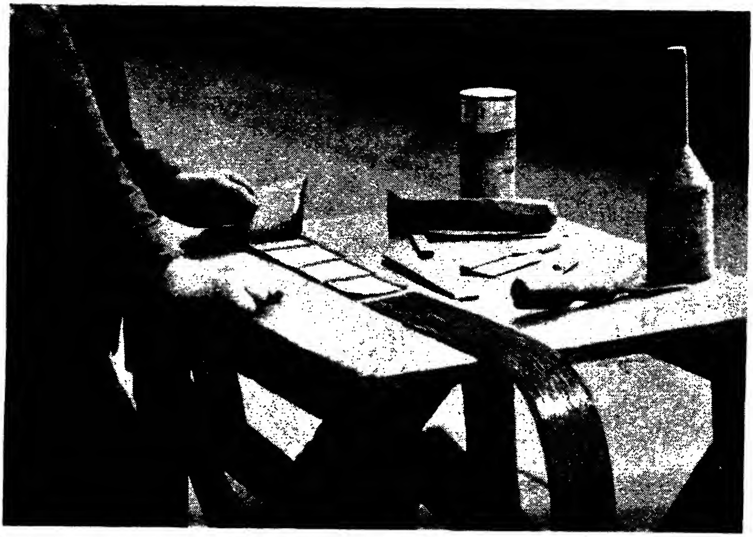


Fig. 4.—Splicing balata belt

Showing how second top ply is cut back to ½ in. of the butt. The worker is holding the longer top ply with his left hand.

Fig. 5.—Splicing balata belt

Pressing the ends in press block after solutioning and heating. Block must first be wetted to prevent belt from sticking to it. Allow belt to remain in press for ten minutes.

the press for about ten minutes. It may then be removed and allowed to cool. The edges can then be carefully pared and rounded over with a warm knife. No ragged edges must remain, and the red covering should be dressed over the raw edges.

Splicing Leather Belts

To make a leather belt splice, the use of a spokeshave is necessary. With this the leather is tapered away from the butt or splice mark to the end of the belt. The rough work is best done with the spokeshave and then trimmed with a scraper and very sharp knife. Another useful tool is a small steel plane. Make sure that the taper is true and gradual, and there must be no lumps behind the butt (Fig. 3).

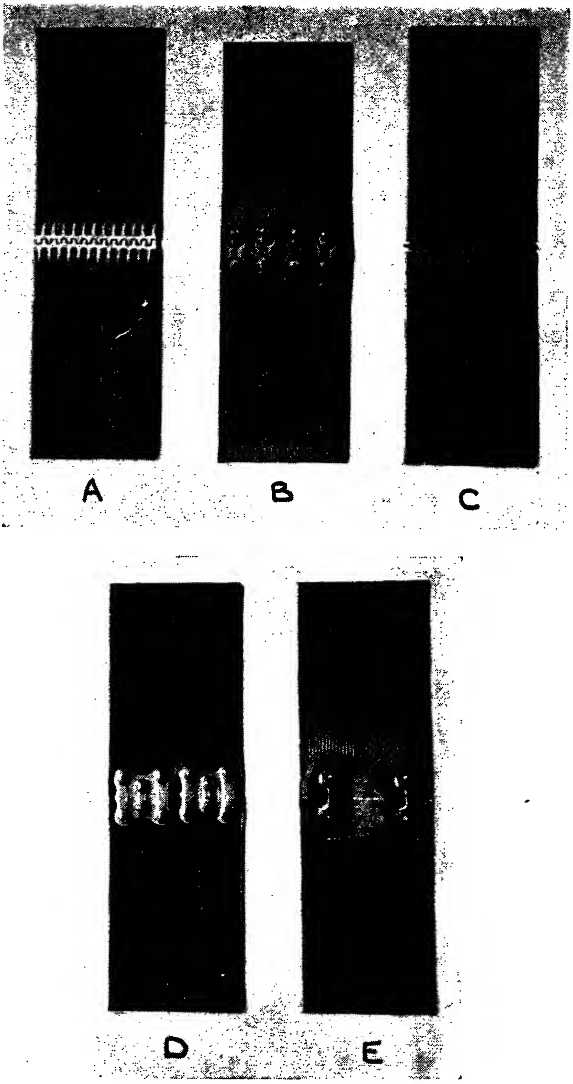


Fig. 6.—Some metal belt fasteners

A, bat fastener; *B*, Jackson button fastener; *C*, clipper fastener; *D*, crescent fastener; *E*, Jackson plate fastener.

Cements and how use them

The glue used must be hot, and spread thinly, working well into the belt. The two ends should then be brought together, and rubbed with a slight pressure, then placed in the clamp or press. This can be made from two boards and a pair of bench clamps. The glue or cement usually sets within an hour.

There are two kinds of belt cements, one for dry and the other for

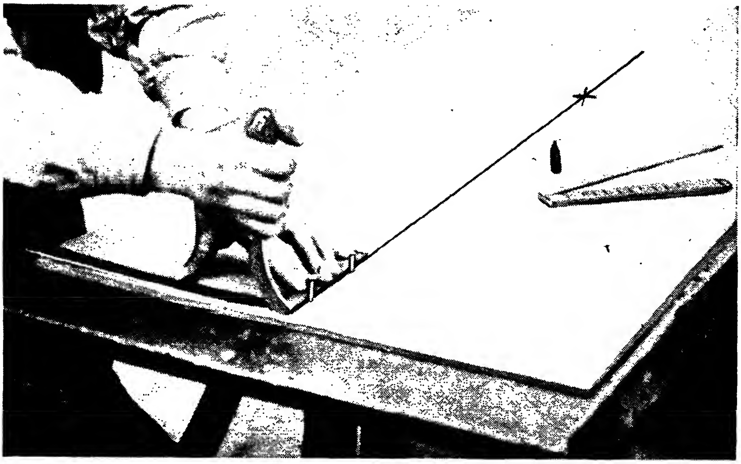


Fig. 7.—Method of cutting broad belts

Using a short metal square. Note small nails on centre line to steady the square.

damp conditions. The first is purchased in cakes or lumps, which dissolve in a glue pot. This cement should be heated, but not boiled, and should be reduced with hot water to the proper consistency to apply easily. It must really be hot and fresh.

The other type is very inflammable and must be kept away from an open light. It is sold ready to apply. Three coats must be given, each coat being allowed to dry before the other is applied. When the third coat is being put on the brush must be worked fast, as it dries quickly. The two ends are then brought together and pressed in the clamp. When well made, such a joint lasts for years, but must be given an occasional inspection.

Length of Lap for Leather Belting

The following table gives the minimum amount of lap required.

Width of Single Belt in inches	*Length of Lap in inches*
1–4½	4½
5	5
6–8	6
9	6½
10–14	7
15–24	8

All double belts should have a 6-in. lap at least.

Metal Fasteners

As regards the many kinds of metal fasteners on the market, little need be said, except that there are types for every kind of service, and if a good brand is selected and the manufacturer's instructions are followed, good service will be given. A few of the most popular kinds of fasteners are shown in Fig. 6.

Cutting the Belt

When cutting a belt for a metal fastener or for a laced butt joint, a metal square should be used. This is to ensure that the cut is at right angles to the centre line of the belt. If a square is not employed, one or both ends are likely to be cut uneven, and these conditions cause the belt to slip from side to side of the pulley and impose irregular strains which will shorten the life of the belt. The knife blade should be made wet occasionally to make it cut easily, in the case of a leather belt.

Fig. 8.—PUNCHING LEATHER BELTS FOR LEATHER LACING

Oval punch holes parallel with belt.

Cutting Wide Belts

Sometimes it is difficult to cut a wide belt square on account of slight variations in the width of the belt, and because the sides may not be parallel. If this is so, the following method has been found satisfactory.

Find the centre of the belt near the point to be cut, then at a point 2 ft. or 3 ft. back from this dimension locate the centre again and between these two points draw a clean sharp line. Next hold the leg of the square against the centre line just made, and trim off one half of the end of the belt by cutting along the other leg.

By laying the square on the other side of the centre line the other half can be cut. The square should be held firmly in position while this is being done, and if two small nails are driven in the belt on the centre line, they will help to keep the square from slipping.

HOW TO LACE A BELT

Punching the Belt

In punching a belt for lacing, it is desirable to use an oval punch, the longer diameter of the punch being parallel with the belt, so as to

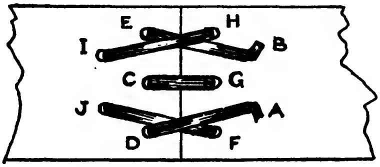


Fig. 9.—BELT LACING

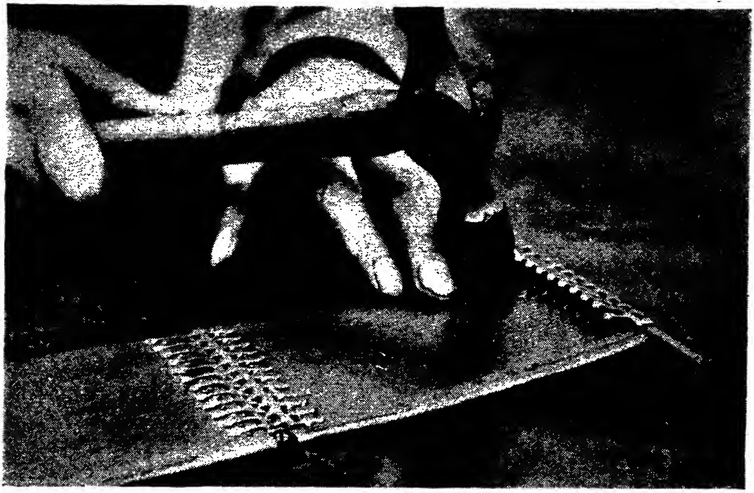


Fig. 10.—Alligator-type fastener being driven home

Hinge gauge pin *A* in use only when applying lacing ; never use as hinge pin. *B* shows sectional steel rocker hinge pin in place. Cut this pin same width as bar on lacing.

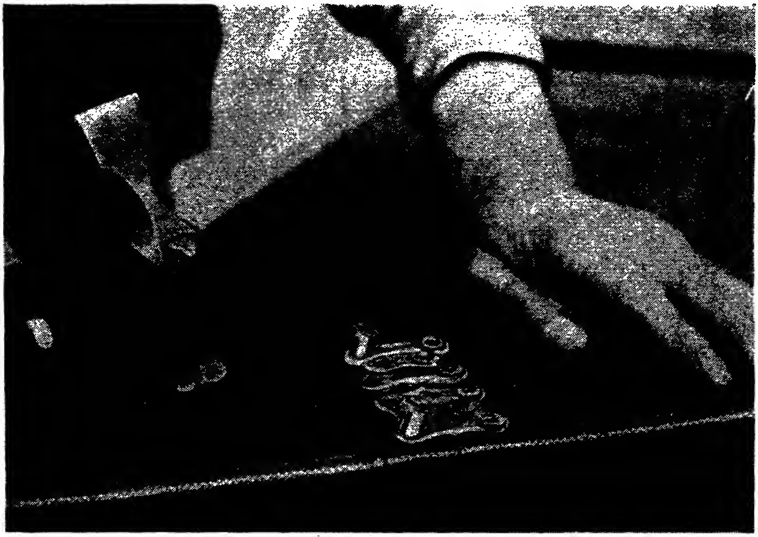


Fig. 11.—Joining belt up with crescent fastener

Showing plates attached to one end of belt. When the other end is in position start the two outside rivets with the prongs slanted away from the joint as shown above. As they are driven in, they draw the ends of the belt together. Drive other rivets in the ordinary manner.

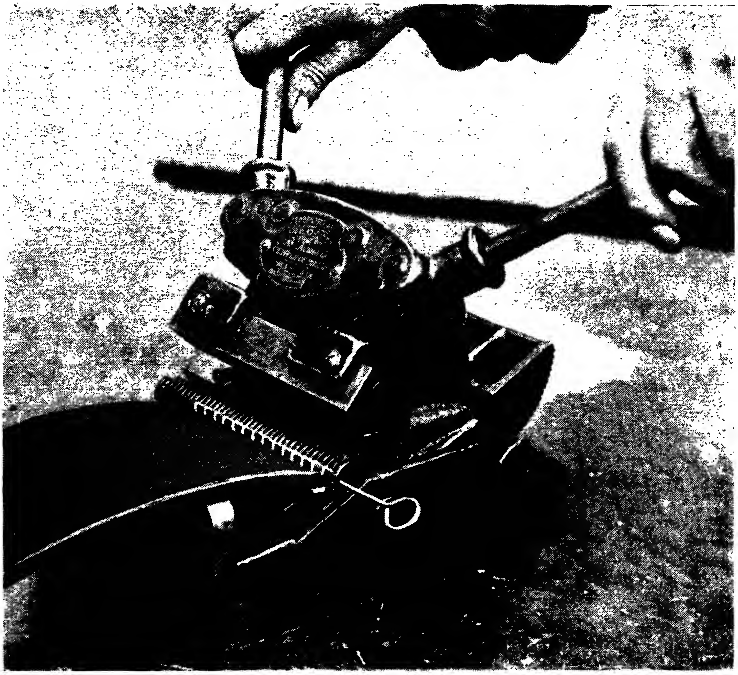


Fig. 12.—CLIPPER FASTENER BEING FIXED BY THE CLIPPER BELT LACER

A row of clipper hooks of the proper width is placed in the tool. The square end to be laced is then inserted, with the gauge pin in position as shown; the lever is then depressed, which imbeds the hooks deeply in the belt, leaving a uniform row of loops. When the other end is similarly treated, the two ends are brought together and a rawhide or fibre pin is slipped through the loops.

cut off as little leather as possible. There should be in the belt two rows of holes as shown in Fig. 8. Begin to lace in the centre of the belt, and take great care to keep the ends exactly in a line and to lace both sides with equal tightness. The lacing should not be crossed on the side of the belt that runs next to the pulley (Fig. 9).

For belts about 3 in. wide and over, the holes *C*, *D*, *E* and *F*, *G*, *H* should be about 1 in. apart and $\frac{5}{8}$ in. from the line of joint; *A*, *B* and *I*, *J* should be about $\frac{1}{2}$ in. behind *C*, *D*, *E*.

Size of Lace

The size of laces for various widths of belting should be as under:

Width of Belt	*Width of Lace*
24 in. and over	$\frac{1}{2}$ in.
6–24 in.	$\frac{3}{8}$,,
2–4 in.	$\frac{5}{16}$,,
2 in. and less	$\frac{1}{4}$,,

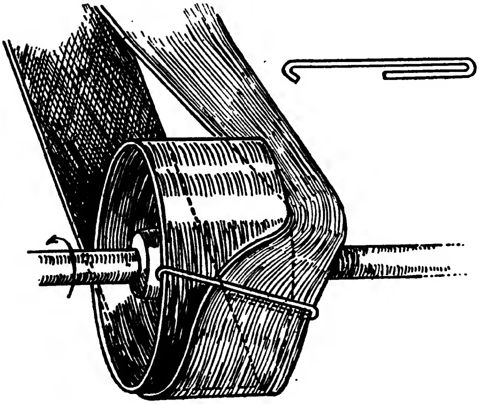


Fig. 13.—A SIMPLE DEVICE FOR PUTTING BELT ON PULLEY

This is a piece of round iron bent to the shape illustrated, which is hooked on pulley. When the engine is started slowly, it will pull belt on to pulley.

Putting on a Heavy Belt

A simple device for putting on heavy belts, when stationary, in place of a rope, is shown in Fig. 13. A piece of round iron $\frac{1}{4}$ in. or $\frac{5}{16}$ in. is bent to shape shown in the small sketch. The short end is hooked over the edge of pulley and the longer end over the belt as shown. Start the motor or engine slowly, and when the belt is on, it is not necessary to stop, as the short end of the hook will straighten out and disengage itself. Use a rod of as light a weight as will answer the purpose, since it will straighten out and free itself with less stress on the belt; the iron hook then falls on the floor.

CARE AND MAINTENANCE OF BELTS

The life of a belt will be prolonged and its driving powers kept at capacity by giving it proper attention.

Keep Belt Clean

Never allow dirt or other matter to collect on the surface of the belt. Such dirt forms into lumps, tearing or distorting the driving face of the belt, and preventing it from forming proper contact with the face of the pulley.

Avoid Oil

Care should be taken to prevent oil from getting at the belt. It will cause loss of power by slip, and is particularly harmful to balata or cotton belts, tending to separate the plies. It will also rot a leather belt and rapidly lessen the efficiency.

An application of ground chalk will absorb the oil on a leather belt and make it workable for a time.

Preventing Chemical Deterioration

In certain unventilated situations, such as bakeries or close to the ceilings of rooms in which a large number of people work, all belts are apt to suffer by a chemical action caused by the bad air. An occasional application of castor oil on both sides of a balata belt, very sparingly put on by means of a rag, will combat deterioration from this cause.

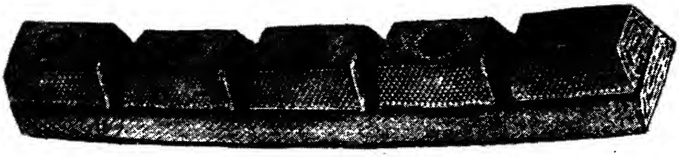


Fig. 14.—SECTION OF V-TYPE BELT ROPE

This example is made with cotton duck impregnated with balata gum, similar to balata belting.

Driving Side of Belt

The canvas side of a balata belt is the driving face, whether the drive be crossed or open.

Opinions differ as to which is the correct driving side of a leather belt, but the balance appears to be in favour of the grain side. In the case of a good leather belt the grain side should transmit nearly twice the power conveyed by the flesh side. The flesh side, which has the greatest tensile strength, will stand the stretching strain necessary in the outside bend around the pulleys. The flesh side, being the stronger, must be protected from wear, and any portion of this being removed will greatly reduce its wearing qualities.

Keeping Leather Belt in Order

To keep a leather belt from becoming dry and lifeless, it should be given an occasional washing and brushing in warm water, and afterwards a good greasing with pure beef tallow or with one of the reliable dressings which are sold for this purpose.

Rubber Belts

Rubber belts should never be run on to pulleys, particularly wide belts.

For endless belts, the best method is to take out both shafts, place the belt in position on the pulleys, and replace the shafts in their bearings.

Maximum Speed of Rubber Belts

The maximum speed at which a rubber belt should be operated is 5,000 feet per minute. Greater life and economy are obtained where the speed is kept inside the 4,000 ft. per minute limit.

Overloaded

If a belt is overloaded, two belts running loose, one on the top of the other, called a rider belt, will increase the power by about 50 per cent. They should, however, not be run on small-diameter pulleys.

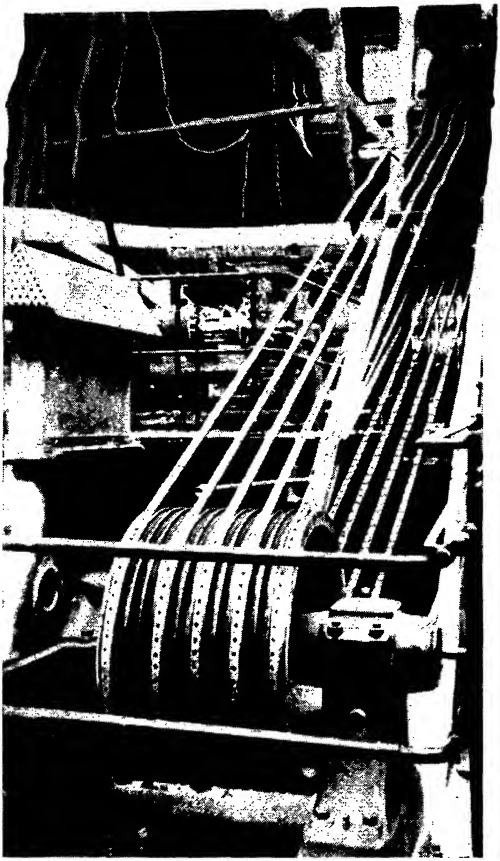


Fig. 15.—A V-ROPE DRIVE

Five $1\frac{1}{8}$-in. wide ropes running at 6,000 ft. per minute.

Avoid using crossed belts on short-centre drives.

Avoid the crossing of wide belts. If it is necessary to cross belts, see that the rubbing of the belt surfaces does not tend to open up the laps.

In order to obtain a given amount of power, it is better to use a narrow belt running at a high speed than a wide belt running at a slow speed. If possible, use a belt speed of 3,000 ft. per minute for smaller powers, and about 4,000 ft. per minute for high powers.

Width of Pulley and Crown

Use a pulley that is wide enough so that a small amount of irregularity does not make it run off. Allow at least $\frac{1}{2}$ in. of machine pulleys and 1 in. or more on countershaft and line-shaft pulleys. Use crowned pulleys wherever possible: $\frac{1}{16}$–$\frac{3}{32}$ in. per foot of pulley width is considered good practice. The camber of the driving and driven pulleys must be the same.

When forks are necessary, they should be of the roller type. The proportion between the diameters of two pulleys working together should not exceed 6 to 1.

Any drive between the horizontal and 45° gives good service. Avoid vertical drives.

Never use mineral oils or resin as a dressing.

Wood and Cast-iron Pulleys

The slip in the case of a wood pulley with a single-ply oak-tanned leather belt is less than one of cast iron, if the belt tension is low. With the same belt tension $33\frac{1}{3}$ per cent. more power can be transmitted with a wood pulley than with either cast iron or steel, so that to transmit the same power the belt on the wood pulley need not be so tight, and in consequence the strain on the bearings is relieved.

BELT TROUBLES—CAUSES AND CURES

If Belt runs badly

The alignment of pulleys and/or shaft may be faulty.

One or both of the pulleys may be badly balanced.

If steam is coming from the engine irregularly, a wave motion will be imparted to the slack side of the belt.

A heated bearing—the result of failure to lubricate—will have this effect.

An overloaded belt will cause it to slide from side to side on the pulley.

If Belt slips

Slip is evidenced by the pulley faces becoming polished. In the case of a balata belt a few drops of castor oil applied to the face of one of the pulleys or to the driving face of the belt by means of a rag will probably cure this. It is better to use one of the recognised dressings on the market for leather belting.

If the belt is still found to be slipping, it should be shortened.

Slip may also be due to the belt being overloaded; a comparison of the actual conditions with the horse-power tables will show if this is the case.

The belt may be too heavy for the small pulley.

The pulley centres may be too short, belt contact with the pulleys being lost thereby.

If a longer drive cannot be arranged, a jockey pulley or gravity idler may be installed with advantage.

The belts may be too narrow for their work. A broad, thin belt is always preferable to a narrow, thick belt.

If Belt leaves the Pulley

It may be running at an excessively high speed.

The alignment of the pulleys may be at fault.

Joint may not have been cut square.

Belt may be overloaded.

If Belt wears or frays at the Edges

Flanged pulleys may be the cause; they serve no useful purpose and should be discarded.

If the shifter is being forced over instead of being eased over, it will not only wear the edges, but buckle the belt. Use roller forks wherever possible.

If metal fasteners are used to join a belt running on a fork drive, they will strike and barb the forks if they are too close to the edge of the belt. The forks in turn will destroy the edge of the belt.

Loss of Power

A belt put on the pulleys too tight causes unnecessary strain on the bearings, and also on the belt. It will develop less power than if put on at the proper tension. Generally speaking, the more slack in a belt up to slipping point, the more power will be transmitted.

This does not apply in all cases, however, as certain drives require a tight belt.

Chapter III

NOTES ON V-ROPE DRIVE

IN spite of the very high efficiency of cotton-rope drives, attempts were made, towards the end of the nineteenth century, to modify the rope section in order to increase its power capacity. The horse-power capacity of even large-diameter cotton ropes in relatively small, and for large main drives, transmitting, say, 2,000–3,000 h.p., from forty to sixty ropes would be necessary, thus necessitating heavy and cumbersome drives. The section adopted finally as an alternative to the round cotton rope was the so-called V, although actually the shape is trapezoidal. The earlier V ropes were found to be not very satisfactory, and have been superseded by the more recent balata V ropes.

THE JOINTED V ROPE

The " Dixel " element driving V ropes consists of a moulded V or trapezoidal shaped belt of cotton duck impregnated with balata in plies. Attached to the narrow face of the belt is a number of elements made of the same material. The elements are tapered on the sides in conjunction with the belt to fit the pulley groove, and also at the ends for clearance when the rope is flexed round the pulley. The angle of the rope is 45°, although the angle of the groove is not necessarily the same, as this varies with the pulley diameters and speeds.

Metal fasteners may be used for making the joint, but a spliced joint is to be preferred. For splicing, the elements must be removed for a sufficient length to give a good joint ; this varies from 12 in. to 18 in. according to section, and the ends scarfed. Care must be taken that the rivet holes coincide, and the ends must then be heated up to 150° F. and covered with solution and placed in a press for about ten minutes. The elements must then be replaced and the joint will be quite uniform with the rope.

" Lincona " V Balata Rope

The " Lincona " V balata rope is a laminated rope made up of sections of balata belting and built up in such a way as to form a V rope. The balata belting is made of cotton duck and balata gum, and in this respect is similar to the " element rope " described above. The layers of balata belting are strongly riveted together and solutioned, thus forming a very strong and flexible unit. For the Lincona rope the joint is a special staggered splice (Fig. 2) made uniform with the rope, and the ropes are despatched with the ends prepared for splicing, the design of which must

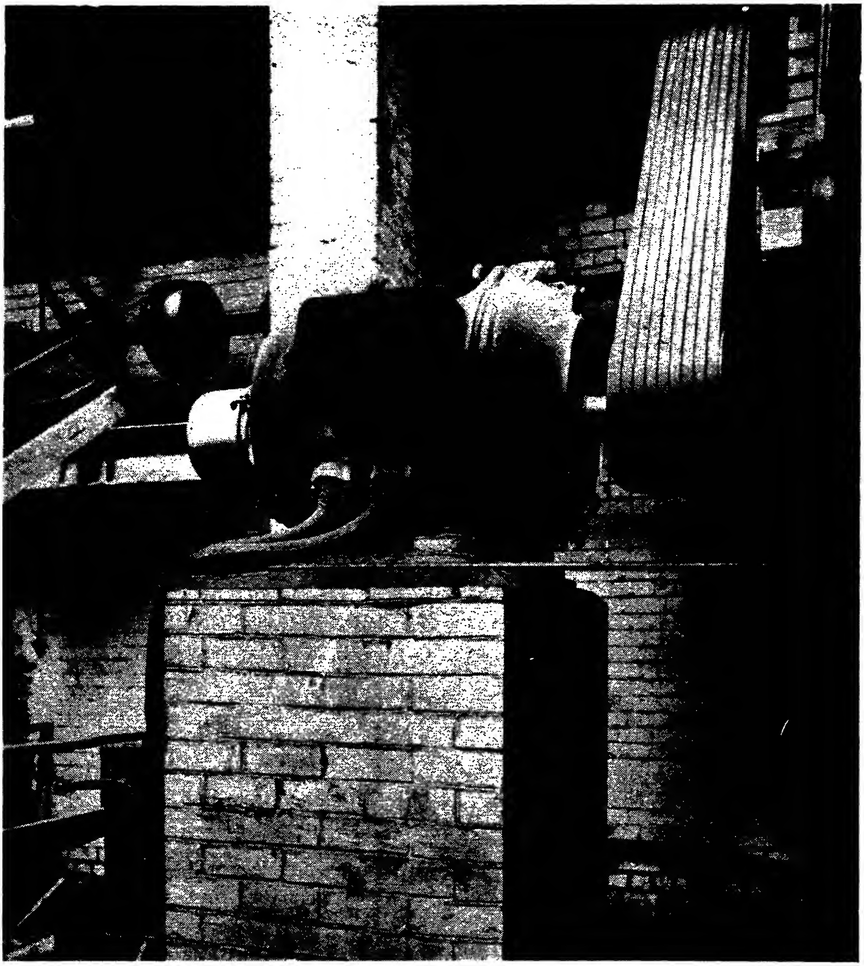


Fig. 1.—TYPICAL EXAMPLE OF V-BELT DRIVE
Showing a 100-h.p. motor driving a line shaft to a brickworks. (*BTH*)

not be altered in any way. A handy-man can then fix the splice in a few minutes. The angle of both the Lincona rope and groove is 35°.

Both the ropes considered are very strong and flexible when compared with the round cotton rope. They are capable of about twice the power, and can be used on smaller pulleys than equivalent cotton-rope sizes.

Effective Torque

The wedging action of the V ropes in the pulley grooves gives this type a great advantage over the round cotton rope. A further advantage is

RECOMMENDED GROOVE ANGLES FOR "DIXEL" ROPES

Size of Rope	*Speeds up to* 3000 *F.P.M.*		*Speeds over* 3000 *F.P.M.*	
¾ in.	6–12 in. pulley	above 12 in. diam.	6–12 in. pulley	above 12 in. diam.
	32°	35°	30°	34°
⅞ in.	8–16 in. pulley	above 16 in. diam.	8–16 in. pulley	above 16 in. diam.
	32°	40°	30°	38°
1 in.	12–18 in. pulley	above 18 in. diam.	12–18 in. pulley	above 18 in. diam.
	34°	38°	32°	36°
1⅛ in.	16–24 in. pulley	above 24 in. diam.	16–24 in. pulley	above 24 in. diam.
	34°	38°	32°	38°
1¼ in.	16–24 in. pulley	above 24 in. diam.	16–24 in. pulley	above 24 in. diam.
	36°	40°	34°	38°
1⅜ in.	20–30 in. pulley	above 30 in. diam.	20–30 in. pulley	above 30 in. diam.
	40°	42°	38°	40°
1½ in.	20–30 in. pulley	above 30 in. diam.	20–30 in. pulley	above 30 in. diam
	42°	45°	40°	42°
1⅝ in.	24–36 in. pulley	above 36 in. diam.	24–36 in. pulley	above 35 in. diam.
	38°	40°	36°	38°
1¾ in.	24–36 in. pulley	above 36 in. diam.	24–36 in. pulley	above 36 in. diam.
	40°	42°	38°	40°
1⅞ in.	30–40 in. pulley	above 40 in. diam.	30–40 in. pulley	above 40 in. diam.
	40°	42°	38°	40°
2 in.	30–40 in. pulley	above 40 in. diam.	30–40 in. pulley	above 40 in. diam.
	42°	45°	40°	42°
2⅛ in.	36–44 in. pulley	above 44 in. diam.	36–44 pulley in.	above 44 in. diam.
	42°	45°	40°	42°
2¼ in.	42–50 in. pulley	above 50 in. diam.	42–50 in. pulley	above 50 in. diam.
	36°	40°	34°	38°

The foregoing table of groove angles is given as a general guide for normal drives.

Any extreme conditions of high speeds, small pulleys, short centres, and large ratios may alter the given angle. In such cases, the makers should be consulted.

The minimum centres required for each size of rope is as follows :

Speed up to 3000 f.p.m., centres = the sum diameters of both pulleys + 50 per cent.
For speeds above 3,000 f.p.m., centres = the sum diameters of both pulleys.

Keep high-speed drives as near the minimum as possible. This overcomes whip of the rope. It is necessary to over-rope short centres, but short length of rope compensates for this.

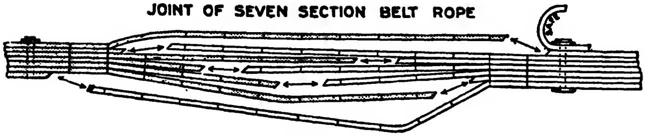
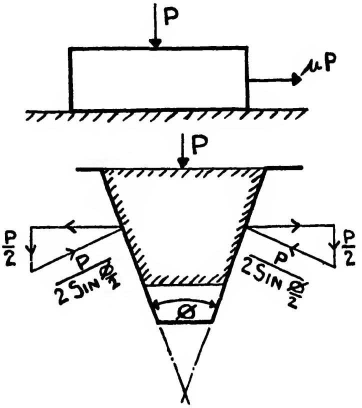


Fig. 2.—Method of jointing "Lincona" rope

the relatively large area of contact between sides of the grooves and ropes. The calculations for effective torque transmitted are the same as for flat belt drives, with the exception of the much higher friction value obtained, due to the wedge action of the ropes.

For flat belts the effective torque is given by :

Torque = $(T_1 - T_2)\ R$ when T_1 = tight side tension
T_2 = slack side tension
R = pulley radius.



Coil Friction

The fundamental formula for coil friction is :

$T_1/T_2 = e^{\mu\theta}$ when $e = 2{\cdot}7183$
μ = coefficient of friction between surfaces
θ = angle of lap in radians.

The friction value of any particular pressure P between flat surfaces is equal to $P \times \mu$, and using the same pressure P on a wedge surface, the friction value is $P\mu \times 1/\sin\frac{\varphi}{2}$, because the pressure between the sides of the rope and groove is increased in the ratio of $1/\sin\frac{\varphi}{2}$. The value of $\mu/\sin\frac{\varphi}{2}$ is usually referred to as the "false μ," and is equal to $2{\cdot}6\mu$ and 4μ for groove angles of 45° and 28° respectively. From the above it will be noted that with the V rope less tension is required for a particular torque, which is very good for the ropes and increases the life of them.

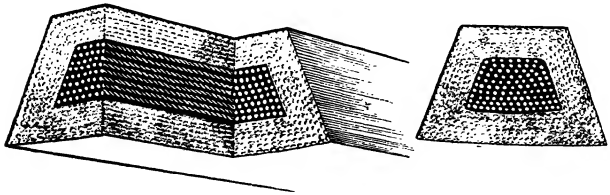


Fig. 3.—Section through V-rope belt

In addition, much shorter centre distance can be adopted and the total pull on the

HORSE-POWER TABLE FOR "DIXEL" ROPES

Speed of Rope in Ft./Min.	¾ *in.*	⅞ *in.*	1 *in.*	1⅛ *in.*	1¼ *in.*	1⅜ *in.*	1½ *in.*	1⅝ *in.*	1¾ *in.*	1⅞ *in.*	2 *in.*	2⅛ *in.*	2¼ *in.*
1,000	3·5	5·5	8·3	10·4	12·5	14·5	17·0	20·1	22·8	25·1	27·8	30·8	33·8
2,000	6·9	11·7	16·3	20·5	24·6	28·5	33·5	39·6	44·9	49·4	54·8	60·7	66·6
2,500	8·5	14·4	20·0	25·2	30·3	35·1	41·2	48·7	55·3	60·8	67·5	74·7	82·0
2,600	8·8	14·9	20·7	26·9	31·4	36·3	42·6	50·4	57·2	62·9	69·9	77·3	83·7
2,800	9·4	15·9	22·1	28·7	33·4	38·7	45·5	53·8	61·1	67·1	74 6	81·4	89·4
3,000	10·0	17·0	23·5	30·5	35·5	41·1	48·4	56·1	65·0	71·3	79·3	86·6	95·1
3,200	10·6	18·0	24·9	32·3	37·6	43·5	51·3	59·4	68·9	75·5	84·0	91·7	100·8
3,400	11·1	18·8	26·0	33·8	39·4	45·6	53·7	62·2	72·1	79·1	88·0	96·1	105·5
3,600	11·6	19·7	27·2	35·3	41·4	47·6	56·1	65·0	75·4	82·7	91·9	100·4	110·3
3,800	12·0	20·4	28·2	36·7	42·7	49·4	57·2	67·6	78·5	87·0	95·9	104·5	115·0
4,000	12·5	21·2	29·3	38·1	44·3	51·3	60·5	70·1	81·2	89·2	99·2	108·2	119·0
4,200	13·1	22·2	30·7	39·9	46·5	53·8	63·4	73·4	85·1	93·4	103·8	113·4	124·5
4,400	13·6	23·1	31·9	41·4	48·3	55·8	65·8	76·2	88·4	96·9	107·8	117·7	129·3
4,600	14·0	23·8	32·9	42·7	49·7	57·5	67·7	78·5	91·0	99·8	111·0	121·2	133·1
4,800	14·3	24·3	33·6	43·6	50·7	58·7	69·2	80·2	92·9	101·9	113·3	123·8	135·8
5,000	14·6	24·8	34·3	44·5	51·8	60·0	70·6	81·9	94·9	104·1	115·7	126·4	138·8
5,200	15·0	25·5	35·2	45·7	53·2	61·6	72·6	84·1	97·5	106·9	118·9	129·9	142·6
5,400	15·3	26·0	35·9	46·6	54·3	62·8	74·0	85·8	99·4	109·0	121·3	132·4	145·5
5,500	15·5	26·3	36·4	47·2	55·0	63·6	75·0	86·9	100·7	110·5	122·9	134·2	147·4

bearings, i.e. $(T_1 + T_2)$ is considerably reduced. The fact that smaller pulleys and fewer ropes at shorter centres can be used makes these drives very compact, thus effecting considerable saving in floor space, etc.

Horse-power

Rope speeds of 3,000–4,000 ft. per minute give the highest efficiency, and drives based on these velocities are the most economical. Higher rope speeds, of course, can be used, and velocities as high as 7,000 ft. per minute have been used with success. The horse-power curve is not a straight line, but a closed curve, the maximum power obtainable being at a velocity of about 6,000–7,000 ft. per minute.

Above this figure the horse-power rapidly decreases, due to the increased centrifugal force, which increases as the square of the velocity. Even above 4,000 ft. per minute, the curve flattens out fairly considerably, which is the reason why this velocity is considered the most ecomonical.

The expression for the horse-power of a V rope is H.P. $= \dfrac{(T_1 - T_2)\,V}{33{,}000}$ and the ratio of T_1 to T_2 is obtained from $e^{\mu\theta} = T_1/T_2$ where μ is the "false μ" mentioned above and V is rope vel. in ft./min. This is not a great deal of use, however, unless the safe load, i.e. T_1 and the coefficient of friction are known. The makers of the ropes issue horse-power tables

HORSE-POWER TABLE FOR "LINCONA" V BALATA ROPES

Speed of Rope in Feet per Min.	*Area of Cross-section in Square Inches*									
	·25	·50	·65	·80	1	1·35	1·75	2	2·25	2·50
1,000	3·2	6·2	8·2	10·1	12·6	17·0	21·9	25·0	28·2	31·3
2,000	6·1	12·4	16·1	19·8	24·7	33·4	43·4	49·5	55·8	62·0
2,500	7·6	15·3	19·9	24·5	30·6	41·3	53·5	61·2	68·8	76·5
2,600	7·9	15·8	20·5	25·2	31·5	42·5	55·0	63·0	71·0	78·9
2,700	8·2	16·2	21·3	26·2	32·7	44·2	57·2	65·5	73·5	81·9
2,800	8·4	16·8	21·9	27·0	33·7	45·5	59·0	67·5	76·0	84·5
2,900	8·7	17·4	22·7	27·9	34·9	47·1	61·0	69·7	78·4	87·1
3,000	8·9	17·9	23·3	28·7	35·8	48·4	62·8	71·6	80·5	89·6
3,100	9·2	18·4	24·1	29·6	37·0	50·0	64·9	74·0	83·1	92·5
3,200	9·5	19·0	24·7	30·4	38·0	51·3	66·2	76·0	85·3	95·0
3,300	9·7	19·4	25·3	31·1	38·9	52·5	68·0	77·8	87·5	97·2
3,400	9·9	19·9	26·0	32·0	40·0	54·0	70·0	80·0	89·8	99·9
3,500	10·2	20·4	26·6	32·7	40·9	55·2	71·5	81·9	92·0	102·2
3,600	10·4	20·9	27·1	33·4	41·7	56·3	73·0	83·5	94·0	104·5
3,700	10·6	21·4	27·8	34·1	42·7	57·2	74·9	85·5	96·2	107·0
3,800	10·9	21·9	28·4	34·9	43·7	59·0	76·5	87·5	98·5	109·3
3,900	11·1	22·2	29·0	35·7	44·6	60·3	77·6	83·9	100·0	111·5
4,000	11·3	22·5	29·3	36·0	45·0	60·8	78·7	90·0	101·2	112·5
4,100	11·5	23·1	30·1	37·1	46·3	62·5	80·8	92·5	104·0	115·9
4,200	11·8	23·5	30·6	37·7	47·1	63·6	82·2	94·0	105·9	117·5
4,300	12·0	24·0	31·2	38·4	48·0	64·7	84·0	96·0	108·0	120·0
4,400	12·2	24·3	31·6	38·9	48·6	65·6	85·0	97·2	109·5	122·0
4,500	12·4	24·6	32·2	39·6	49·5	66·9	86·5	98·8	111·5	123·5
4,600	12·6	25·1	32·6	40·2	50·2	67·7	87·8	100·5	113·0	125·5
4,700	12·7	25·4	33·0	40·6	50·8	68·5	89·0	101·8	114·5	127·0
4,800	12·8	25·7	33·5	41·2	51·5	69·5	90·0	102·9	115·8	128·7
4,900	13·0	26·1	33·9	41·7	52·2	70·5	91·7	104·5	117·9	131·0
5,000	13·1	26·3	34·1	42·0	52·5	71·0	92·0	105·0	118·1	131·3
5,100	13·3	26·6	34·6	42·6	53·2	71·7	93·0	106·5	119·7	133·0
5,200	13·5	27·1	35·2	43·3	54·2	73·2	94·8	108·5	122·0	135·5
5,300	13·7	27·3	35·5	43·7	54·6	73·7	95·2	100·0	122·5	136·3
5,400	13·8	27·5	35·8	44·0	55·0	74·2	96·0	110·0	123·9	137·4
5,500	13·9	27·8	36·3	44·6	55·8	75·4	97·2	111·5	125·2	139·0
5,600	14·0	28·0	36·5	44·9	56·1	75·8	98·0	112·0	126·1	140·1
5,700	14·1	28·2	36·6	45·0	56·3	76·0	98·5	112·7	126·9	140·8
5,800	14·2	28·5	37·0	45·6	57·0	77·0	99·8	114·0	128·3	142·5
5,900	14·3	28·6	37·2	45·8	57·2	77·3	100·0	114·3	128·6	143·0
6,000	14·3	28·7	37·3	45·9	57·4	77·5	100·1	114·8	129·2	143·4

based on 180° arc of contact, and a correction is necessary for all drives having a speed ratio greater than 1 : 1. The loss in arc of contact on small pulleys for ratios greater than 1 : 1 is easily calculated from:

$$\text{Loss in arc of contact in degrees} = \frac{(D - d)\,60}{C}$$

Where D = large pulley diameter in inches
d = small pulley diameter in inches
C = centre distance in inches,

and factors for different losses are as follows:

Loss in Arc in Degrees . . .	0	10	20	30	40	50	60
Multiply Listed H.P. by . .	1	·97	·96	·92	·89	·85	·8

The loss here calculated is based on the assumption that both tight and slack sides take the form of straight lines. In cases where the slack side is on the bottom, the sag will still further reduce the arc of contact.

Obtaining Best Results

For best life and service, pulleys larger than the maker's recommended minimum should be used, and the centre distance should be as long as reasonably convenient. A drive working with minimum-diameter pulleys and shortest centre distance cannot be expected to give as long a life as one using larger pulleys and working at long centres. In these cases the number of ropes should be increased, so as to counteract the extra wear and tear on the ropes.

The normal minimum centre distance should not be less than about two to three times the diameter of the larger pulley, and for shorter centres than this 20–30 per cent. more ropes should be used. A useful criterion for wear and tear on the ropes is the number of trips the ropes make in space per second, five or six being a normal figure.

Thus a high-speed, small-ratio drive using minimum pulleys at the shortest centre distance needs very careful consideration, and for 1 : 1 ratio drives, pulleys considerably larger than the minimum must be used if a reasonable life is expected. Although with the slack side on the bottom the arc of contact on the smaller pulley is reduced, it is considered an advantage for long centre drives to have the tight side on top, as a better balance is then obtained for the rope.

Balata V ropes can be used for quarter twist drives, but they need special consideration and should only be used with small-ratio drives at long centres. Damp and moisture do not affect the ropes, but their greatest enemy is heat. The highest temperature in which these ropes should be used is 100° F. When a correctly fitted drive has been running a week or two, the ropes, which initially should have sufficient tension to keep both slack and tight sides straight, will slacken and form a slight catenary curve on the slack side. If the ropes tend to jump in the grooves, they should be tightened a little.

Although the balata V-rope drive is relatively compact when compared with the round cotton rope drive, it still leaves much to be desired in the saving of floor space, even for main drives. In machine drives, and all cases where very short centre distances are essential, this form of drive is not suitable, and up to recent years mechanical drives had to be used until the advent of the endless V-rope drive.

HORSE-POWER TABLES (SINGLE ROPES)

Velocity in Feet per Min.	*Size of Rope*					*Velocity in Feet per Min.*	*Size of Rope*				
	½ in.	*$\frac{21}{32}$ in.*	*⅞ in.*	*1¼ in.*	*1½ in.*		*½ in.*	*$\frac{21}{32}$ in.*	*⅞ in.*	*1¼ in.*	*1½ in.*
500	0·5	0·7	1·8	3·0	4·0	2,550	2·2	2·8	6·6	12·7	17·3
550	0·5	0·8	2·0	3·3	4·4	2,600	2·2	2·8	6·7	12·9	17·5
600	0·6	0·8	2·2	3·6	4·8	2,650	2·2	2·9	6·8	13·1	17·8
650	0·6	0·9	2·3	3·8	5·0	2,700	2·2	2·9	6·9	13·3	18·0
700	0·7	1·0	2·5	4·2	5·5	2,750	2·3	3·0	7·0	13·5	18·3
750	0·7	1·1	2·7	4·5	6·0	2,800	2·3	3·0	7·1	13·7	18·5
800	0·8	1·1	2·9	4·8	6·4	2,850	2·3	3·1	7·2	13·9	18·8
850	0·8	1·2	3·0	5·1	6·8	2,900	2·3	3·1	7·3	14·1	19·0
900	0·9	1·2	3·0	5·3	7·2	2,950	2·4	3·2	7·4	14·3	19·3
950	0·9	1·2	3·0	5·5	7·5	3,000	2·4	3·2	7·5	14·5	19·5
1,000	0·9	1·2	3·0	5·5	7·5	3,050	2·5	3·3	7·6	14·7	19·8
1,050	1·0	1·3	3·1	5·8	7·9	3,100	2·5	3·3	7·7	14·8	20·0
1,100	1·0	1·3	3·2	6·0	8·2	3,150	2·5	3·4	7·8	15·0	20·3
1,150	1·0	1·4	3·3	6·3	8·6	3,200	2·5	3·4	7·9	15·1	20·5
1,200	1·0	1·4	3·4	6·5	8·9	3,250	2·5	3·5	8·0	15·3	20·8
1,250	1·1	1·5	3·5	6·8	9·3	3,300	2·5	3·5	8·1	15·4	21·0
1,300	1·1	1·5	3·6	7·0	9·6	3,350	2·6	3·6	8·2	15·6	21·2
1,350	1·2	1·6	3·7	7·3	10·0	3,400	2·6	3·6	8·3	15·7	21·4
1,400	1·2	1·6	3·8	7·5	10·3	3,450	2·6	3·7	8·4	15·9	21·6
1,450	1·3	1·7	3·9	7·8	10·7	3,500	2·6	3·7	8·5	16·0	21·8
1,500	1·3	1·7	4·0	8·0	11·0	3,550	2·7	3·8	8·6	16·2	22·0
1,550	1·4	1·8	4·2	8·2	11·3	3,600	2·7	3·8	8·6	16·3	22·2
1,600	1·4	1·8	4·3	8·4	11·6	3,650	2·7	3·9	8·7	16·5	22·4
1,650	1·5	1·9	4·5	8·6	11·9	3,700	2·7	3·9	8·7	16·6	22·6
1,700	1·5	1·9	4·6	8·8	12·2	3,750	2·8	4·0	8·8	16·9	23·0
1,750	1·6	2·0	4·8	9·0	12·5	3,800	2·8	4·0	8·9	17·2	23·3
1,800	1·6	2·1	4·9	9·2	12·8	3,850	2·8	4·1	8·9	17·2	23·3
1,850	1·6	2·2	5·1	9·4	13·1	3,900	2·8	4·1	8·9	17·2	23·3
1,900	1·6	2·2	5·2	9·6	13·4	3,950	2·8	4·2	9·0	17·4	23·4
1,950	1·7	2·3	5·4	9·8	13·7	4,000	2·8	4·2	9·0	17·5	23·5
2,000	1·7	2·3	5·5	10·0	14·0	4,100	2·8	4·2	9·0	17·5	23·5
2,050	1·8	2·4	5·6	10·3	14·4	4,200	2·8	4·2	9·1	17·6	23·5
2,100	1·8	2·4	5·7	10·5	14·8	4,300	2·8	4·2	9·1	17·6	23·5
2,150	1·9	2·5	5·8	10·8	15·0	4,400	2·8	4·2	9·2	17·7	23·5
2,200	1·9	2·5	5·9	11·0	15·2	4,500	2·9	4·3	9·2	17·7	23·5
2,250	1·9	2·6	6·0	11·3	15·5	4,600	2·9	4·3	9·2	17·8	23·5
2,300	1·9	2·6	6·1	11·5	15·8	4,700	2·9	4·3	9·3	17·8	23·5
2,350	2·0	2·7	6·2	11·8	16·1	4,800	2·9	4·5	9·3	17·9	23·5
2,400	2·0	2·7	6·3	12·0	16·4	4,900	2·9	4·5	9·5	17·9	23·5
2,450	2·1	2·8	6·4	12·3	16·7	5,000	3·0	4·5	9·5	18·0	23·5
2,500	2·1	2·8	6·5	12·5	17·0						

MINIMUM PULLEY DIAMETERS

Size of V Rope	½ in.	$\frac{21}{32}$ in.	⅞ in.	1¼ in.	1½ in.
Smallest satisfactory diameters .	3 in.	5 in.	9 in.	13 in.	22 in.
Minimum diameters	2½ in.	4½ in.	8 in.	12 in.	20 in.

ENDLESS V-ROPE DRIVES

The endless V-rope drive is now one of the most popular transmission mediums for motor drives, and it is now agreed that, in spite of theoretical objections, this form of drive gives remarkable efficiency and service. Designed in the first place to meet a long-felt want in the textile industry for a flexible drive, its use has been extended, and it is now used in almost every industry, and is giving equal satisfaction on the lightest laboratory testing machine and heavy pump and rock-crushing machinery absorbing up to 1,000 h.p.

The section is trapezoidal in shape, having an included angle of 40°, which gives a " false μ " equal to 3μ. It is of rubber-cotton construction, thus having a very high coefficient of friction and flexibility. Expensive moulds are required for its manufacture, which ensure all the ropes of a particular size being identical in section and length. The load-carrying core is made of an endless cotton cord, which is impregnated with rubber and then vulcanised under pressure in the moulds. The strands of the cords are so arranged that each is isolated from its neighbour by rubber, thus preventing them from rubbing against each other and causing internal breakdown.

Horse-power Values

Calculations for horse-power and T_1 and T_2 values for the endless V rope are the same as given above for the jointed V rope, although different values must be used for φ and μ.

Rope Length

As the endless V rope is made in standard lengths, care must be taken in calculating rope length and pulley diameters for the desired centre distance. Once the pulley diameters and centre distance are fixed for a particular drive, the rope length required is given by :

$$L = \frac{\pi}{2}(D + d) + 2C + \frac{(D - d)^2}{4C}$$

Where L = pitch length of rope in inches
D = diameter of large pulley in inches
d = diameter of small pulley in inches
C = centre distance in inches.

Except for trial-and-error purposes this formula is not very useful, as the problem in practice is to find, either the centre distance when pulley diameter and rope length are known, or the pulley diameters when rope length and centre distance are known. By far the more usual case is,

POWER FACTOR TABLE

Application	*Percentage H.P. extra*
Line shafts from line starting motors	0% to 10%
Cotton-ginning machines	0% to 20%
Machine tools	0% to 30%
Printing machines	0% to 30%
Mining Machinery	0% to 40%
Laundry Machines	10% to 20%
Coal-mining Machines	10% to 40%
Reversing Motors	20%
Flour and feed-mill machines	20% to 30%
Pumps and Compressors	20% to 30%
Woodworking machines	20% to 30%
Pulp and paper-mill machines	20% to 40%
Oil-field machinery	20% to 50%
Rock-crushing machines	20% to 50%
Textile machines	20% to 50%
Brick and clay machines	25% to 50%

when pulley diameters and rope length have been decided upon, to find the exact centre distances from the following :

$$C = \sqrt{\frac{C_1^2}{4} - \frac{(D-d)^2}{8}} + \frac{C_1}{2}$$

Where C = required centre distance in inches
D and d = pulley diameters as above
$C_1 = \frac{1}{2}\left[L - (D+d)\frac{\pi}{2}\right]$
L = pitch length of rope as above.

This formula is not as frightening as it looks and will be found very useful once it has been mastered. (See also practical notes on fitting belts.)

The unique construction of these ropes makes them particularly suitable for high-speed ratios and small pulleys. As they depend only upon elastic tension for driving, i.e. they do not rely on their own weight, very short centre distances are permissible, and the pull on the bearing is relatively small when compared with joined V, cotton rope, and belt drives even with small pulleys and arc contacts of 120°. The endless construction enables them to give very high efficiencies at high rope speeds, and velocities of 5,000 and even 6,000 ft. per minute are common. A slight resiliency makes them ideally suitable for pulsating loads, and for certain textile machines, where a quiet getaway without an end-breaking snatch is essential, the endless V rope is considered a necessary part of the machine.

As with the jointed V rope the formula for horse-power, viz. $H.P. = \frac{(T_1 - T_2)\,V}{33{,}000}$, is not very useful unless full details of the friction coefficient and safe load are known and the maker's horse-power lists should be consulted. The rubber-cotton construction of the rope gives it a very high coefficient of friction, and laboratory tests give a T_1/T_2 ratio of 16, although the calculated ratio is over 100. This figure, i.e. 16, compares very favourably with both cotton and jointed V ropes, which have respective T_1/T_2 ratios of 7 and 10, and gives a very efficient performance figure, for maximum power is thus transmitted for any given tight-side tension with minimum pull on the bearings.

Centre Distances

Even the largest endless V rope, it will be noted from page 605, will not transmit a power comparable with cotton or jointed V ropes, and therefore for very large powers the width of the pulleys, due to the large number of ropes, tends to become rather cumbersome. It is rare, however, that very large power drives require very short centre distances, and the jointed V-rope drive is usually adopted. Whereas the normal minimum centre distance for a jointed V-rope drive is approximately equal to twice or three times the diameter of the larger pulley, with the endless V-rope drive the minimum centre is equal, for speed ratios of 3 : 1 or over, to the difference in pulley diameters.

This difference in permissible centre distances is fairly considerable, as will be seen from the following example : for a drive with 18-in. and 70-in. diameter pulleys the minimum centre distances for the jointed and endless V-rope drives are 140 in./210 in., and 52 in. respectively.

For an endless V-rope drive with pulleys less than 3:1 ratio, the minimum centre distance is that at which the pulley rims foul. As these ropes depend only on elastic tension for driving and work at much shorter centres than the jointed type, it is possible to predict accurately exactly what power they will transmit, and for this reason a much lower factor of safety is used.

It is an advantage to be able to calculate just what power a V-rope drive will transmit, although it must be understood that because of this and the relatively low factor of safety used, care must be taken in designing drives for other than steady loads and the arc of contact on the smaller pulley must be taken into account.

The horse-power list gives the powers the different sizes will transmit at various speeds, assuming 180° arc of contact on the small pulley. For speed ratios other than 1 : 1 a correction is necessary for loss in arc of contact. A further correction for type of machine is then necessary, and all manufacturers' catalogues give power-factor tables similar to that shown on page 58.

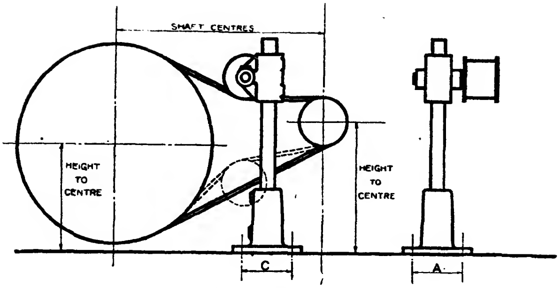


Fig. 4.—V-ROPE-DRIVE TENSION DEVICES FOR FIXED CENTRES—VERTICAL-PILLAR TYPE

Providing the arc of contact on the small pulley is not less than 120°, there seems to be no limit to pulley ratio, and drives giving up to 14 and 15 : 1 speed reduction have been installed. A useful rule to remember, for drives with speed ratios 3 : 1 or over, is that when the centre distance is equal to the difference in the pulley diameters, the arc of contact in the smaller pulley is 120°, and therefore for this range of speed ratios the minimum centre distance for particular pulley diameters is easily calculated. A correctly designed drive will run without bodily slip, although there will be a speed loss of approximately 1 per cent., due to the so-called "creep," which is caused by the difference in tight- and slack-side tensions acting on a resilient rope. In determining the pulley diameters for a particular speed ratio, the driving pulley should be made 1 per cent. greater in diameter in order to give the desired driven speed.

Uses for Endless V-rope Drive

The wide range of adaptation of the endless V rope is impressive. Unaffected bytemperatures between minus 40° and plus 140° F., and steam, moisture, or water, they can be used with success in conditions far worse than normal and for which it would be difficult to find an alternative. Proof that dusty and gritty atmospheres are not detrimental is found in the large number of drives on coal atritors and grinders of every type. The two great enemies of the standard-quality V rope are temperatures above 160° F. and any kind of oil, although ropes made with synthetic rubber can now be obtained which will give a reasonable life in temperatures up to 200° F. and when running in oil.

In certain industries, particularly in flour mills and aluminium-crushing plants, long-rope and flat-belt drives are extremely dangerous, due to the presence of static electricity. If care is not taken efficiently to

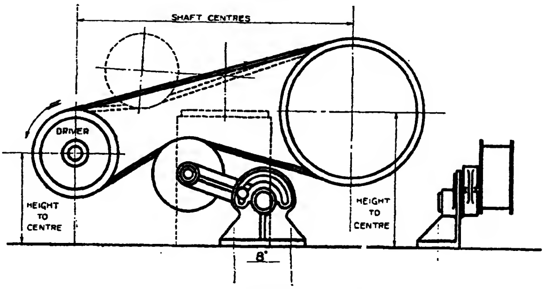


Fig. 5.—V-ROPE-DRIVE TENSION DEVICES FOR FIXED CENTRES—SWING-ARM TYPE earth the driven machine, sufficient potential may be built up between the rope or belt and machine so as to cause a flash. Owing to the fact that there is no bodily slip in the endless V-rope drive, and the much smaller volume of each V rope, the tendency to build up statics is much less than in the other form of drive. In addition, the sharp corners of the trapezoidal section are conducive to rapid dissipation of static, so that the endless V rope has been found quite safe in applications of this kind.

Adjustment of Centres

As the ropes are endless, adjustment of centres is necessary in order to keep the correct tension in the ropes. The majority of drives are from electric motors, and the standard slide rails supplied with these machines form an excellent means for tensioning the ropes. Judging whether or not the ropes are under correct tension is rather difficult for the inexperienced. In practice they should have just sufficient tension to give the correct driven speed when under full load. If bodily slip is taking place, the smaller pulley will soon become very warm and the ropes should be tightened until the correct driven speed is again obtained.

If the centres are fixed, a jockey pulley must be used. The spring or weight-loaded type as used for belt drives is entirely unsuitable for endless V ropes, and the design generally adopted is one in which the jockey pulley can be easily adjusted and nipped up into the desired fixed position.

There are two or three general standard arrangements (Figs. 4 and 5), which can be designed to suit almost any layout. Usually a flat double-flanged belt pulley is used for the jockey, although whether or not this can be used depends entirely on the position of it. If the jockey pulley cannot, due to structural obstruction, be placed in the correct position, a " finned " pulley may be necessary in order to keep the ropes in position.

Generally a flat double-flanged pulley is suitable when placed near the driving pulley on the slack side, and should not be more than a third of the centre distance away from the point where the ropes leave this pulley. All jockey pulleys should be placed on the slack side of the drive where the tension is very small, so that the bearings carrying the pulley are not overloaded. It is always advisable for fixed centre drives to be submitted to the makers, so that the drive can be carefully considered on its merits.

Points to Remember

A correctly designed drive, whether jointed or endless V, can be guaranteed to give excellent service, but it must be remembered that usually makers depend entirely upon the user for full information of working conditions and starting and peak loads, etc., and a drive designed on inadequate information cannot be expected to give the best service. This applies particularly to the endless V-rope drive, where the factor of safety is not so great as that used for jointed V ropes. The tendency to effect a slight saving in initial cost by reducing the number of ropes to a minimum cannot be too strongly condemned. It has been proved conclusively that an extra rope increases the life of the drive by a far greater proportion than the percentage increase in cost.

The life of the endless V-rope drive must necessarily be much shorter than the jointed type because the wear and tear in the endless ropes is much greater, due to the high speeds for which they are generally used and the small pulley diameters and short centre distances. Against this, however, the cost of replacements is relatively small and they are easily fixed, and the saving in floor space more than counteracts this.

Chapter IV

FRICTION CLUTCHES

WHEN it is desired to connect a stationary shaft to one which is revolving, or to stop a pulley from revolving without bringing the driving shaft to rest, some kind of clutch must be used.

" Jaw "- or " Claw "-type Clutches

In the early days of engineering this was accomplished by " jaw "- or " claw "-type clutches (Fig. 1), but as this type of clutch is instantaneous in action it is not suitable for operation when the shafts are running, as this would cause excessive stresses which would result in breakage of either shaft or clutch. Hence the need soon arose for a gradual method of starting.

The Friction Clamp

Fig. 2 shows a simple device which was an early attempt to solve the problem of instantaneous starting, and consists of a friction clamp with " claw " clutch. The friction clamp will permit a little slipping at the moment of going into gear, and whilst this reduces the shock at starting, it can be a source of trouble due to slipping when driving full load, and is also difficult to release when transmitting heavy powers. Engineers, therefore, attempted to find means to overcome these difficulties.

Essential Requirements of a Friction Clutch

A correctly designed friction clutch should enable a rotating shaft or pulley to be connected to a stationary shaft, and should gradually accelerate the latter to the speed of the driving member. Further, it must enable the driven member to be disconnected at will from the driving member, without having to reduce the speed of the driver. Finally, it should be possible to run the driving member with the driven member stationary for such periods of time as desired without undue heating or " drag " taking place.

In other words, it should give complete control of power, and enable the operator of a machine or workroom to have the same control over his motive power as he would if a separate motor were installed.

A modern friction clutch correctly installed will fulfil these conditions, and has become recognised as a convenient and economical unit for controlling power between prime mover and driven machine.

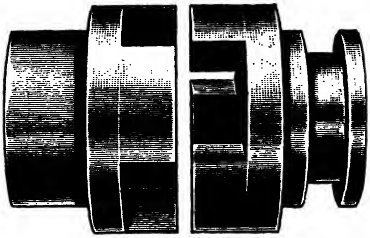


Fig. 1.—An early type of clutch, the " jaw " or " claw " type

The arrangement, erection, and adjustment of the clutch has a great deal to do with the actual results obtained, but the days of troublesome friction clutches are over, and no person need be apprehensive about installing one, providing complete and accurate information is given to the manufacturer.

In order to assist buyers as far as possible to give this information, almost all manufacturers of repute issue clutch inquiry forms.

The most important questions are :

What is the maximum power a clutch must start from rest ?

How frequently will clutch be operated ?

How long will clutch remain disengaged with driving portion revolving ?

Types of Clutches

The type of clutch to adopt is generally decided by the particular purpose for which it is to be used.

Friction clutches may take the form of couplings for connecting two separate shafts, or they may be combined with pulleys or gearwheels, chain sprockets, etc.

It should be clearly understood that the standard design of clutch coupling is not intended to take the place of a flexible coupling, or to act as a centralising device to accommodate varying levels or alignments, unless specially designed for such conditions.

Shaft Bearings

It is most important that shafting which is to be connected by friction clutches should have bearings close up on each side, capable of

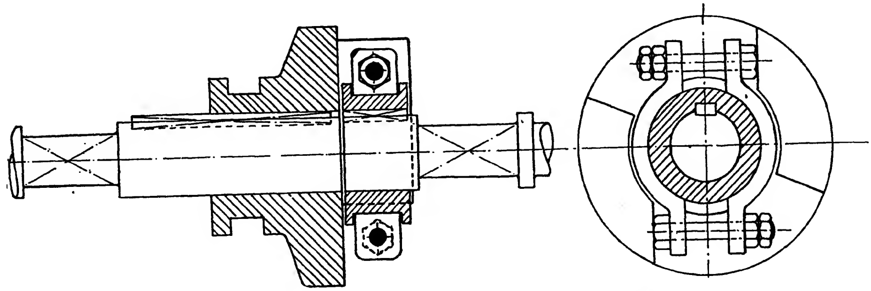


Fig. 2.—A simple device for solving the problem of instantaneous starting

This consists of a friction clamp with a " claw " clutch.

withstanding the temporary end thrust which occurs when the clutch is put in and out of operation. They should be mounted on rigid supports capable of maintaining the shafts in correct level and alignment. If any difficulties are experienced with clutch couplings, the shafts should be checked for accuracy in this respect.

Friction clutches are marketed in sizes which are calculated to transmit certain maximum powers at given speeds when engaged on loads of a regular or uniform character, and working under favourable conditions.

Any departure from the above conditions, say, for example, irregular or shock loads, such as operate with compressors, pumps, calenders, explosive engines, grinding machinery, etc., or treating of materials which introduce great resistance to motion or the continuously starting up of machinery, or the picking up of loads at high speeds, etc., demands special consideration, and the suitable size and type of clutch to be adopted can be determined only when the most exacting conditions with which the clutch may have to contend have been clearly defined.

Clutch Size and Design

In addition to the normal horse-power to be transmitted there are other factors to be considered when deciding clutch size and design.

Nature of the Load

The first of these is the nature of the load. It is obvious that a more robust clutch will be required to drive a 30-h.p. compressor than a 30-h.p. generator.

Torque or Twisting Moment

Another point to be borne in mind is the actual torque or twisting moment to be exerted by the clutch when accelerating the driven machine to full speed. As this is the most severe duty a clutch can be called upon to perform, it should not be underestimated; the following example should make this clear:

A certain design of mixing mill requires approximately 9 h.p. to drive when running at full speed, but the total horse-power for which the clutch should be designed is made up thus:

Driving load to do actual work, including friction in the machine	9 h.p.
Resistance due to "setting" of materials in pan	4½ ,,
Power required to accelerate heavy rotating part of machine up to speed	7½ ,,
Minimum horse-power for which clutch must be designed	21

The higher the speed of the clutch the more important it is to make correct estimates of the horse-power required for acceleration, etc.

Clutches used for continuously picking up heavy loads, especially at high speeds, should have large reserves of power, otherwise there will be rapid generation of heat, wear on linings, and other undesirable contingencies.

H.P. Tables for Various Types of Clutches

We give below horse-power tables for various types of clutches, and recommend the use of the following factors in connection with them :

Type of Drive	*Factor*
Generator drives, Centrifugal pumps, etc.	Multiply h.p. required by 1·25
General millwright work	Multiply h.p. required by 1·5
Compressors	Multiply h.p. required by not less than 3, depending on type, etc.

Example.—If a clutch is required to drive a factory lineshaft requiring 30 h.p., we could select from the tables any design *within the speed limits* given which is listed to transmit not less than 45 h.p. at the particular speed at which the clutch must run.

Owing to the generation of heat, clutches should not be expected to *start up from rest* more than a specified load, and the higher the speed the more important this becomes, as will be seen by reference to tables.

Before the introduction of friction linings, as we know them to-day, friction clutches were designed with friction surfaces of cast iron working against cast iron, with a certain amount of oil and grease on the contact surfaces. These gave reasonably good results at low speeds or where small loads in comparison with clutch size had to be started from rest, but where the starting loads were heavy and slipping periods more than a few seconds, excessive wear took place, giving rise to a whole host of troubles.

In an effort to overcome these difficulties, wood, leather, cork, copper, and bronze were used in connection with cast iron, with varying degrees of success, but unless the friction surfaces can be *adequately* lubricated it is better to use one of the many friction linings which are at present sold for the purpose.

Two Types of Friction Clutches

There are now two distinct types of friction clutches :

(1) Where an endeavour is made adequately to lubricate the friction surfaces (these are generally used on machine tools).

(2) Where it is intended that the friction surfaces shall be kept dry (these are used on general millwright work).

CONE CLUTCH

One of the earliest and simplest of all friction clutches is the cone type (Fig. 3), which transmits power due to the force exerted on the friction surfaces by the internal portion being thrust into the outer.

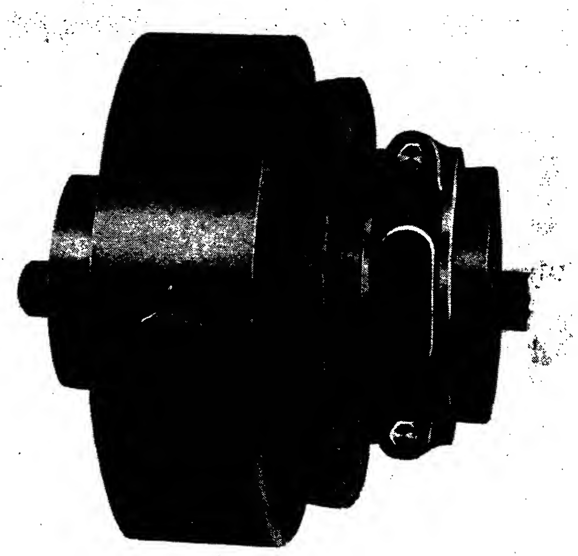


Fig. 3.—CONE-TYPE CLUTCH

This transmits power due to the forces exerted on the friction surfaces by the internal portion being thrust into the outer.

There are still many at work and giving good service, but they have the disadvantage of requiring continuous end pressure upon them all the time power is being transmitted.

Unless proper provision is made for withstanding this continuous end thrust, both in the clutch and on the shaft upon which it is mounted, trouble is bound to occur. Further, the operating gear must be designed to maintain this pressure on the clutch, otherwise slipping will take place.

Cone Angle

If the angle of the internal portion is made small with the intention of reducing the end thrust, the cones are liable to stick, and very great force is required to separate them. On the other hand, if the cone angle is too large the pressure required to keep the two cones together is so great that it is difficult to transmit a reasonable power for the size of clutch. In practice, the angle varies from 8° to 20°, depending upon the materials used for the contact surfaces.

Generally, cone clutches have been used on cranes, winches, etc., where continuous applied pressure during brief periods of use is not objectionable, but they are not to be recommended for even the lightest of ordinary shafting drives.

The table on the following page gives some idea of the proportion and horse-power rating.

The cone-type clutch, whilst being far in advance of the "jaw" type, is very "fierce" in action, and many means have been tried to overcome this, the principal usually being to introduce springs under the lining, etc.

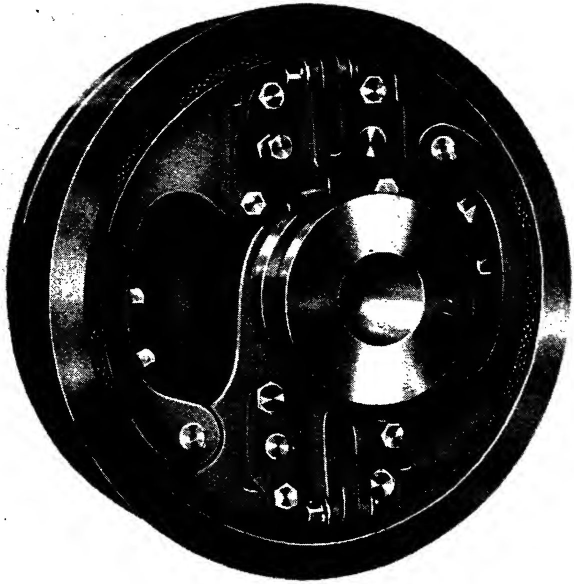


Fig. 4.—An internal-expanding clutch

As other types of clutches have now been perfected, however, it is usual to adopt them in preference to cone clutches where a smooth pick-up is desired.

Thus there are at present on the market numerous types and designs of friction clutches, each possessing some particular mechanical characteristic for engaging the friction surfaces of one portion with those of the other. We illustrate some well-known friction-clutch designs, and state the features of each which make them suitable for the particular duties upon which they are employed.

Overall diameter	*Overall length*	H.P. at 100 r.p.m.	
		With friction lining	*Metal to metal*
12 in.	8¾ in.	2½	1·0
14 in.	9¾ in.	4	1·5
16 in.	11 in.	5	2
18 in.	12 in.	7½	3
20 in.	13 in.	10	4
24 in.	15¼ in.	16	6

INTERNAL-EXPANDING CLUTCH

A number of internal-expanding clutches are on the market and giving good service at ordinary lineshaft speeds.

The actual mechanism varies with different makes. Figs. 4, 5, and 6 show one well-known type.

The clutch proper consists of two main parts, the clutch mechanism, comprising two cast-iron slippers lined with best-quality bonded asbestos, and the external shell.

The two slippers, pivoted on steel fulcrum pins, are expanded and contracted by means of right- and left-hand square-thread screws working in bronze nuts.

The slippers, operating screws, and nuts can be withdrawn for inspection, relining, etc., without interfering with any other part of the installation.

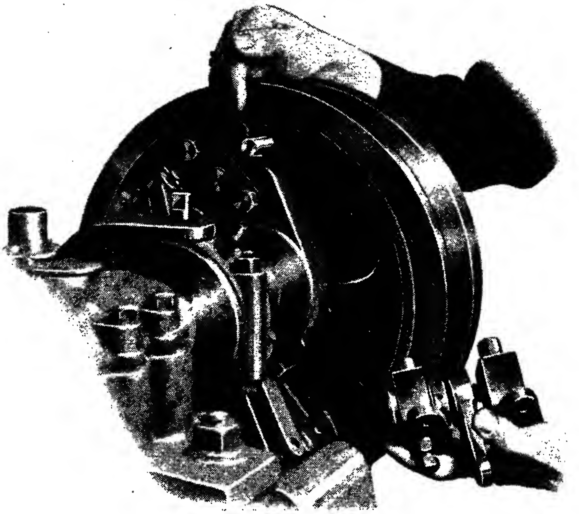


Fig. 5.—An internal-expanding clutch showing the two cast-iron slippers

The position of the sliding collar *N* when in gear for driving should be noted; it is intended to touch the clutch hub.

When in this position the links *R* are "locked" and no force is required to keep the clutch engaged.

Adjustment

Bring sliding collar *N* to "out-of-gear" position, slacken locking screws *L*, and insert tommy-bar in the adjusting holes in brass trunnion nuts *T*.

To increase driving power: Turn tommy-bar inwards towards the

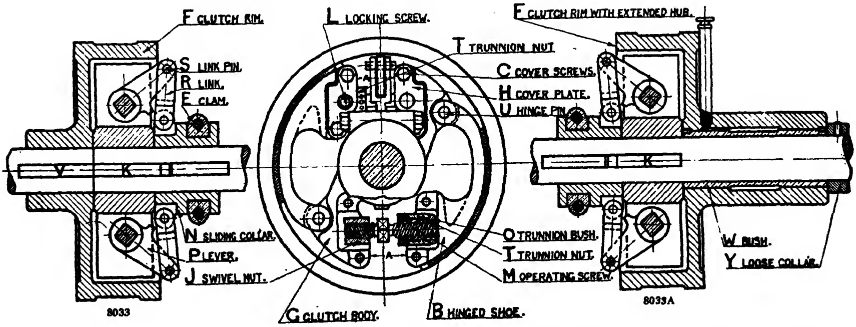

F CLUTCH RIM.
S LINK PIN.
R LINK.
E CLAM.
V
K
H
N SLIDING COLLAR.
P LEVER.
J SWIVEL NUT.
8033
L LOCKING SCREW.
A
G CLUTCH BODY.
B HINGED SHOE.
F CLUTCH RIM WITH EXTENDED HUB.
T TRUNNION NUT
C COVER SCREWS.
H COVER PLATE.
U HINGE PIN.
O TRUNNION BUSH.
T TRUNNION NUT.
M OPERATING SCREW.
W BUSH.
Y LOOSE COLLAR.
8033A

Fig. 6.—Diagram showing the operation of the internal-expanding clutch

INTERNAL-EXPANDING CLUTCHES

HORSE-POWER AND SPEED TABLE

Size of clutch (Diam. in inches)	*Load*	*Revolutions per minute*				
		50	100	200	300	400
10½	S	2½	5	10	15	18
	T	2½	5	10	15	20
13	S	6	12	24	35	40
	T	6	12	24	36	48
16	S	10	20	40	55	
	T	10	20	40	60	
18	S	18	35	70	84	
	T	18	35	70	105	
20	S	20	40	80	99	
	T	20	40	80	120	
24	S	33	65	119		
	T	33	65	130		
28	S	45	90	150		
	T	45	90	180		
37	S	85	170			
	T	85	170			
43	S	115	215			
	T	115	230			
49	S	150	265			
	T	150	300			
61	S	200	310			
	T	200	390			

S = equivalent h.p. which clutch will start up from rest.

T = h.p. which clutch will transmit when full speed has been attained.

Maximum speeds shown by thick lines.

shaft; when the required adjustment has been given, tighten locking screw *L before* bringing sliding collar into "in-gear" position.

The clutch should now pick up the load without shock or undue slip and transmit the power satisfactorily.

Maintenance

When any design of clutch is new, more attention will be necessary to adjustment than when the friction surfaces have become bedded in; afterwards, adjustment will be required only at long intervals.

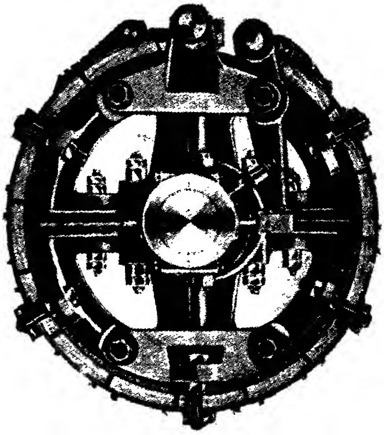


Fig. 7.—A BAND-TYPE CLUTCH

Clutches which are used for frequent stopping, starting, and picking up heavy loads will, of course, require more frequent adjustment, and it is desirable to give periodical attention to this to avoid undue slipping.

How to Withdraw Slippers

To withdraw slippers for examination, etc., take out cover screws *C* and link pins *S* from clutch body, remove cover plate *H*, after which toggle levers *P*, square-threaded screws *M*, and nuts *J* and *O* can be withdrawn as a single unit. Afterwards remove grub-screws which secure fulcrum pins at hinged end of slippers, then withdraw fulcrum pins *U* (tapped hole is to facilitate this).

Slippers can now be removed.

A horse-power table for this design of clutch is given on the previous page.

BAND-TYPE CLUTCH

The external or band clutch is made in many forms. A single band, similar to a brake strap, gripping a central drum and operated by a lever, is shown in Fig. 7. Alternatively the band can be made in two pieces.

Due to the fact that full advantage is taken of "coil effect" this type of clutch is easy to operate.

All the parts are of robust construction, and can be repaired by any ordinary mechanic. In some cases the band is lined with wood, so that the clutch is suitable for dredgers and mining work; in many instances they are at work far away from an up-to-date engineer's workshop.

The band clutch is, of course, suitable only for slow-speed work, as it would be affected by centrifugal force if run at more than normal speeds. Unless specially designed it should run only in one direction.

As it is more expensive to produce than the more modern type of

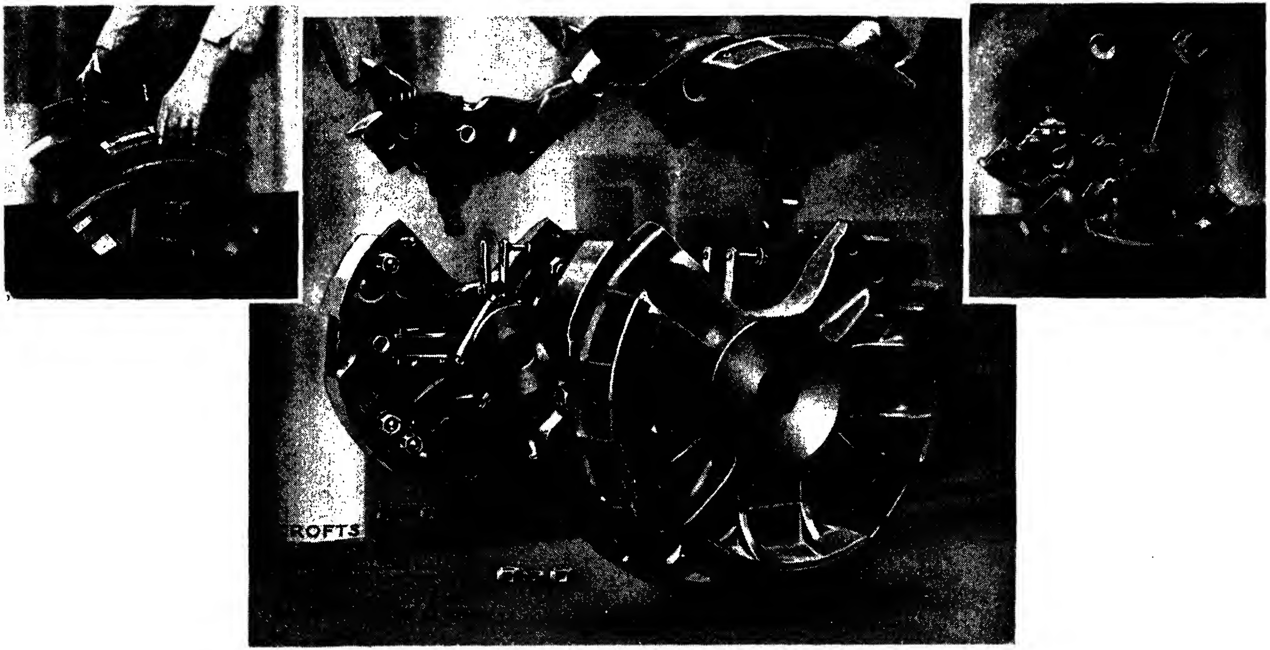


Fig. 8.—A DETACHABLE SLIDE RIM FRICTION CLUTCH
Showing method of removing slide and slippers from clutch.

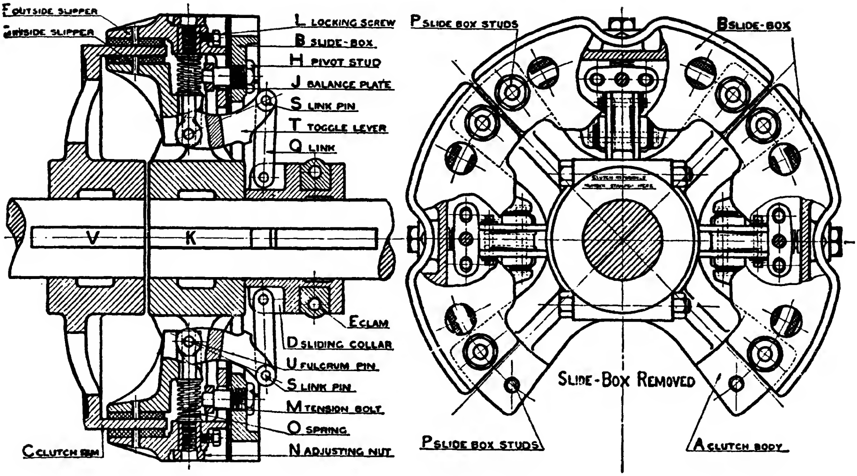

F OUTSIDE SLIPPER
G INSIDE SLIPPER
L LOCKING SCREW
B SLIDE-BOX
H PIVOT STUD
J BALANCE PLATE
S LINK PIN
T TOGGLE LEVER
Q LINK
V
K
E CLAM
D SLIDING COLLAR
U FULCRUM PIN
S LINK PIN
M TENSION BOLT
O SPRING
N ADJUSTING NUT
C CLUTCH RIM
P SLIDE BOX STUDS
B SLIDE-BOX
SLIDE-BOX REMOVED
P SLIDE BOX STUDS
A CLUTCH BODY

Fig. 9.—Diagram showing the construction of a rim-type clutch

clutch (Fig. 9), it is used only where ease of operation and ease of repair are the first considerations.

Band clutches are also made having up to six complete turns around a chilled cast-iron drum, and clutches to transmit 500 h.p. at 20 r.p.m. have been made in this design.

RIM-TYPE CLUTCH

For ordinary millwright work there is no better design of clutch than the rim type.

This is really a modification of an American design known as the Hill type, but many improvements have been incorporated, and the design shown in Fig. 9 can be considered as one of the best yet produced.

For ordinary commercial speeds it is ideal; the action resembles a number of vice jaws, which grip the rim simultaneously and with equal force. The pressure is exerted through the toggle levers and links shown.

The clutch is designed to run with dry friction surfaces, i.e. cast-iron rim and bonded asbestos-lined slippers.

There is large and even clearance between driving and driven portions when the clutch is out of engagement.

The driving force is transmitted through machine slides of large area, and these project beyond the rim to give adequate support to the slippers.

The complete mechanism can be dismantled without moving the clutch body (*see* Fig. 8).

RIM-TYPE CLUTCHES, HORSE-POWER AND SPEED TABLE

Size of clutch (Diam. in inches)	*Load*	*Revolutions per minute.*					
		50	100	200	300	400	500
10	S	6	12	24	36	36	36
	T	6	12	24	36	46	52
12	S	9	18	36	52	49	49
	T	9	18	36	54	72	81
15	S	17	33	66	75	75	
	T	17	33	66	99	126	
18	S	26	52	95	95		
	T	26	52	104	154		
21	S	38	76	140	140		
	T	38	76	152	218		
24	S	60	120	180	160		
	T	60	120	240	325		
27	S	90	180	240			
	T	90	180	360			
30	S	120	240	260			
	T	120	240	470			
36	S	200	320	280			
	T	200	400	750			
42	S	320	480				
	T	320	640				
48	S	450	550				
	T	450	900				
54	S	600	700				
	T	600	1,200				
60	S	775	850				
	T	775	1,550				

S = equivalent h.p. which clutch will start up from rest.
T = h.p. which clutch will transmit when full speed has been attained.
Maximum speeds shown by thick lines.

Adjustment

When correctly adjusted the sliding collar should touch the clutch body when in the "in-gear" position; the link *Q* is then "locked" and the clutch self-sustaining.

There is no tendency to engage when out of gear.

It is simple to adjust, the sliding collar being placed up to the clutch body, nuts *N* tightened up tight and locked by locking screw *L*, and without further trial it is certain to transmit its rated horse-power.

Another point to be noted is the self-centering quality of the rim clutch when in gear for driving. No special provision is, therefore, necessary to centralise the shafts when the clutch is used as a coupling.

Maintenance

Slippers and slipper guides should be lubricated at regular intervals through oil-holes provided.

To remove slide *B*, complete with slippers, from clutch body:

Withdraw link pin *S*, remove nuts and spring washers from studs *P*; the slide box can then be withdrawn and placed on bench for ready removal of slippers.

To remove slippers:

Remove pivot stud *H*; outer and inner slippers *F* and *G* can then be withdrawn for cleaning or relining.

Always fit new linings to both inner and outer slippers when relining.

DISC CLUTCHES

Some engineers prefer disc clutches on account of the smooth external appearance, although many consider this type gives a more gradual start than other designs.

As a type it is made in a variety of forms, not all of which have given satisfactory results.

Trouble in the past has been caused through insufficient clearance between driving and driven surfaces, wear on the driving bolts or feathers, and a tendency to "drag" when disengaged.

Disc clutches are not self-centering, and a gunmetal bush or roller bearing should be fitted (*see* Fig. 11), unless the shafts are abnormally large in comparison with the clutch diameter.

With the design shown the friction surfaces are pressed together by the special design of lever *T*, and the special feature of this clutch is the single-point adjustment.

The illustration shown has only one plate, but the clutch is made with up to four plates if necessary.

Adjustment for Driving

To increase driving power: put sliding collar in "out-of-gear" position, screw back locking pin *L*, and rotate screwed adjusting collar *N* in a clockwise direction until locking pin *L* can be screwed into next slot in body *A*.

Repeat this operation until the clutch will drive the load without slipping.

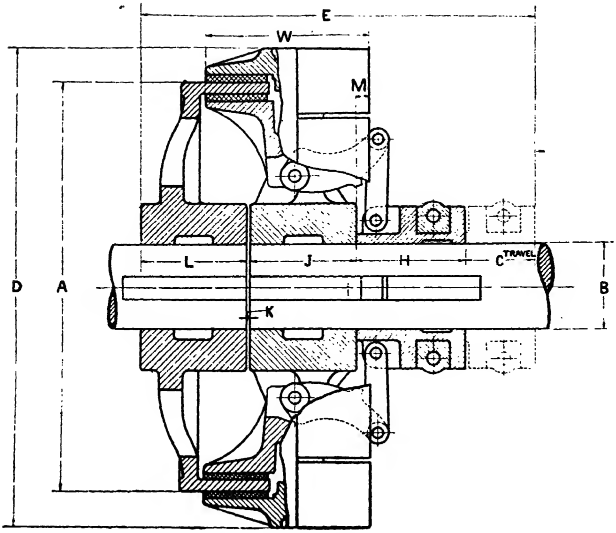


Fig. 10.—A RIM CLUTCH

This illustration shows only one plate, but the clutch can be made with up to four plates if necessary.

Maintenance

Unnecessary slipping should be avoided, as this causes heating up and excessive wear of discs and may cause breakages.

Size of Clutch	*Maximum Shaft Diameter*	*Overall Diameter*	*Maximum Overall Length*	*Distance between Shafts*	*Width of Rim Hub*	*Width of Clutch Hub*	*Width of Sliding Collar*	*Travel of Sliding Collar*	*End of Hub to Back of Clutch Body*	*Overall Width to Clutch Body and Slipper*
A	*B*	*D*	*E*	*K*	*L*	*J*	*H*	*C*	*M*	*W*
10	2½	12½	14	⅛	3½	3¼	5⅛	2	⅜	5½
12	3	15½	15⅝	⅛	4	3⅝	5⅜	2½	9/16	6 5/16
15	3½	18½	16⅝	⅛	4¼	3¾	5¾	2¾	⅝	6 9/16
18	4	21¾	18½	⅛	4¾	4¾	5¾	3⅛	⅝	7 5/16
21	5	25	20⅛	⅛	5¼	5¼	6¼	3¼	½	8
24	6	28	22⅛	⅛	5¾	5¾	6¾	3¾	⅝	8 15/16
27	8	31	24⅝	⅛	6¼	6¼	7¾	4	1 3/16	10½
30	9	35	26	⅛	7	6¾	7¾	4⅜	1 1/16	11⅜
36	10	43	29⅞	⅛	8	7½	9¼	5	1¼	12 9/16
42	10½	50	34½	⅛	9½	8½	10⅜	6	1⅜	14¼
48	11	58	38¾	⅛	11	10	10⅝	7	1⅛	16
54	12	65	44	⅛	13	12	10⅝	8¼	1	18⅜
60	14	72	49¼	⅛	15	14	10⅝	9½	1¼	21⅛

All dimensions in inches.

H.P. TABLE FOR SINGLE-PLATE DISC CLUTCH.

(See Figs. 11 and 12).

Size of Clutch	Max. Shaft Dia.	Load	Revolutions per Minute																				
			100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	1,300	1,400	1,500	1,600	1,700	1,800	1,900	2,000	2,500
in.	*in.*																						
4¼	⅞	S	·6	1·2	1·8	2·4	3·0	3·6	4·2	4·8	5·4	6·0	6·6	6·0	6·0	6·0	6·0	6·0	6·0	6·0	6·0	6·0	6·0
		T	·6	1·2	1·8	2·4	3·0	3·6	4·2	4·8	5·4	6·0	6·6	7·2	7·8	8·4	9·0	9·6	10·2	10·8	11·4	12·0	15·0
5	1¼	S	·8	1·6	2·4	3·2	4·0	4·8	5·6	6·4	7·2	8·0	8·0	8·0	8·0	8·0	8·0	8·0	8·0	8·0	8·0	8·0	8·0
		T	·8	1·6	2·4	3·2	4·0	4·8	5·6	6·4	7·2	8·0	8·8	9·6	10·4	11·2	12·0	12·8	13·6	14·4	15·2	16·0	20·0
6	1½	S	1·6	3·2	4·8	6·4	8·0	9·6	11·2	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	12·0	
		T	1·6	3·2	4·8	6·4	8·0	9·6	11·2	13·0	14·5	16·0	17·5	19·0	21·0	22·5	24·0	25·0	26·0	27·0	28·0	29·0	
8	2	S	3·8	7·6	11·4	15·2	19·0	22·8	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0	24·0		
		T	3·8	7·6	11·4	15·2	19·0	22·8	26·5	30·5	34·0	38·0	42·0	44·5	48·5	52·5	56·0	60·0	64·0	68·0	72·0		
10	2¼	S	8·0	16·0	24·0	32·0	40·0	42·0	42·0	42·0	42·0	42·0	42·0	42·0	42·0	42·0							
		T	8·0	16·0	24·0	32·0	40·0	48·0	56·0	64·0	72·0	80·0	88·0	96·0	104·0	112·0							
12	2½	S	13·0	26·0	39·0	52·0	54·0	54·0	54·0	54·0	54·0	54·0	54·0	54·0									
		T	13·0	26·0	39·0	52·0	65·0	78·0	91·0	104·0	117·0	130·0	143·0	156·0									
14	3¼	S	21·0	42·0	63·0	70·0	70·0	70·0	70·0	70·0	70·0	70·0											
		T	21·0	42·0	63·0	84·0	105·0	126·0	147·0	168·0	189·0	210·0											
16½	3½	S	34·0	68·0	102·0	105·0	105·0	105·0	105·0	105·0	105·0												
		T	34·0	68·0	102·0	136·0	170·0	204·0	238·0	272·0	306·0												
19	4	S	52·0	104·0	110·0	110·0	110·0	110·0	110·0	110·0													
		T	52·0	104·0	156·0	208·0	260·0	312·0	364·0	416·0													
21½	4½	S	75·0	150·0	160·0	160·0	160·0	160·0	160·0														
		T	75·0	150·0	225·0	300·0	375·0	450·0	525·0														

The above table applies also to Coupling and Extended Hub Clutches.

S = equivalent H.P. which clutch will start up from rest.

T = H.P. which clutch will transmit when full speed has been attained.

Maximum speeds shown by thick lines.

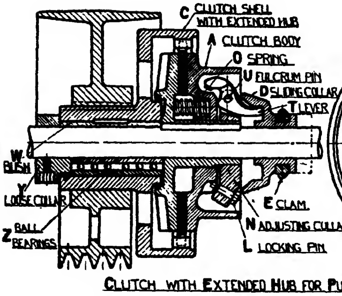

C CLUTCH SHELL WITH EXTENDED HUB
A CLUTCH BODY
O SPRING
U FULCRUM PIN
D SLIDING COLLAR
T LEVER
W BUSH
Y LOOSE COLLAR
Z BALL BEARINGS
E CLAM
N ADJUSTING COLLAR
L LOCKING PIN
CLUTCH WITH EXTENDED HUB FOR PULLEY

Fig. 11.—CLUTCH WITH EXTENDED HUB FOR PULLEY

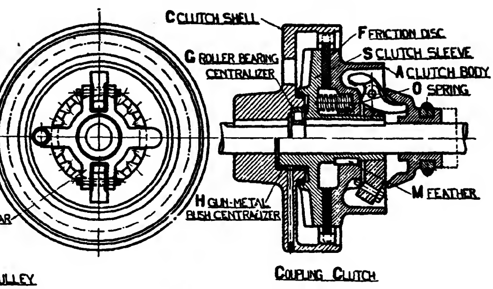

C CLUTCH SHELL
G ROLLER BEARING CENTRALIZER
F FRICTION DISC
S CLUTCH SLEEVE
A CLUTCH BODY
O SPRING
H GUN-METAL BUSH CENTRALIZER
M FEATHER
COUPLING CLUTCH

Fig. 12.—COUPLING CLUTCH

Should a clutch show sign of over-heating, adjustment is urgently needed, but before adjustment is made the clutch should be allowed to cool down.

MULTIPLE-DISC CLUTCH

An interesting type of clutch is made by Messrs. Crofts (Engineers) Ltd., which entirely eliminates levers and pins, the action of the clutch being such that moving the sliding collar to the " in-gear " position forces steel balls down an inclined surface and exerts pressure on driving and driven plates.

When the sliding collar is moved to the " out-of-gear " position, the balls are forced up the inclined surface by means of springs.

The design of sliding collar and angle of cone allows the maximum clearance between driving and driven plates.

When the clutch is used in gear-boxes, etc., it is usual to allow a portion of same to run in the oil or to pour oil over the clutch to lubricate the friction surfaces.

Both driving and driven plates are of hardened steel.

The clutch can be supplied with special-quality fabric discs if it is to work without oil being poured over it.

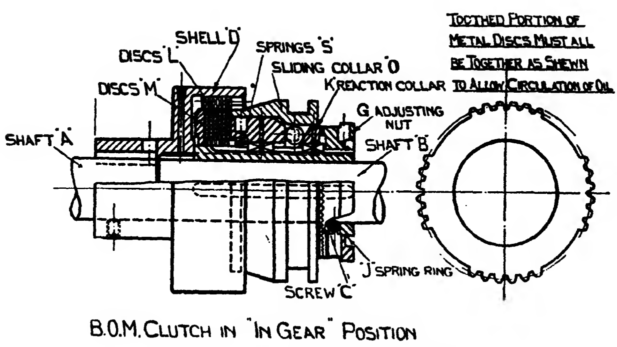

SHELL 'D'
DISCS 'L'
DISCS 'M'
SPRINGS 'S'
SLIDING COLLAR 'O'
K REACTION COLLAR
TOOTHED PORTION OF METAL DISCS MUST ALL BE TOGETHER AS SHEWN TO ALLOW CIRCULATION OF OIL
G ADJUSTING NUT
SHAFT 'A'
SHAFT 'B'
'J' SPRING RING
SCREW 'C'
B.O.M. CLUTCH IN "IN GEAR" POSITION

Fig. 13.—MULTIPLE-DISC CLUTCH

Adjustment (*see* Figs. 13 and 14)

To increase driving power the adjusting nut *G* must be rotated clockwise when looking at the clutch from the nut end.

First put the clutch in the out-of-gear position, then, holding clutch mechanism with shaft *B* or by turning tommy-bar inserted in pressure distance-piece, insert second tommy-bar in adjusting nut *G*, and by a smart blow with hand move the nut one notch at a time. Test for pressure after each notch adjustment.

The clutch can be supplied with one sliding portion to operate two clutches (Fig. 15).

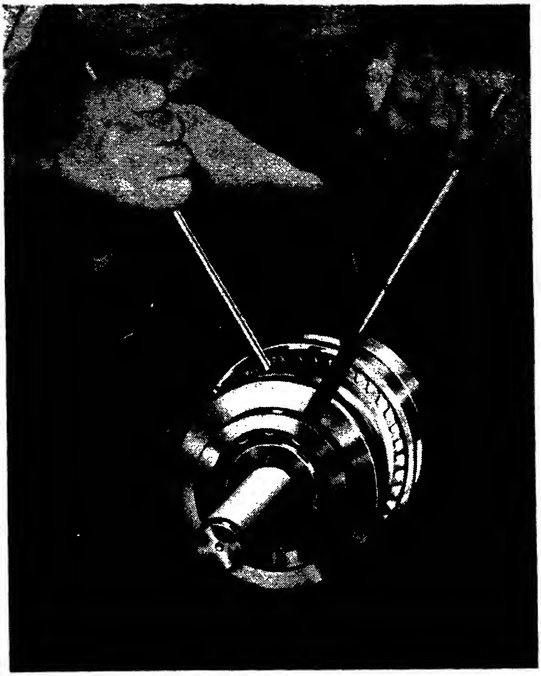


Fig. 14.—Adjusting a multiple disc clutch to obtain increased driving power

INSTALLATION OF FRICTION CLUTCHES

Having now described the general forms which friction clutches take, we will consider the manner in which they are installed.

The most common arrangement is the clutch coupling, which is used to connect two shaft ends (Fig. 17).

Wherever possible, the mechanism portion should be placed on the driven shaft so that when the clutch is disengaged, no lubrication whatever is required.

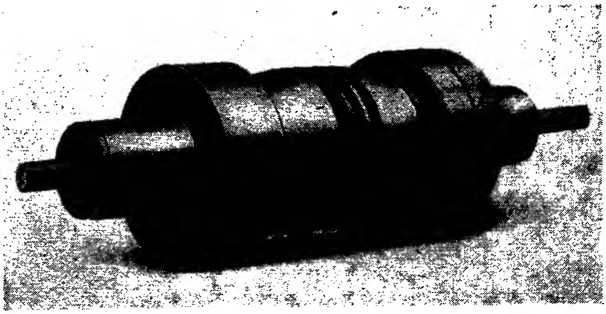


Fig. 15.—Ball-operated multiple disc clutch

This arrangement is strongly recommended, as there are no lubrication difficulties ; the only point to be observed is that the shafts must be set and maintained in correct level and alignment ; where this is not

EXTENDED HUB CLUTCHES (with hub mounted on Gunmetal Bush)

Dimension Diagram

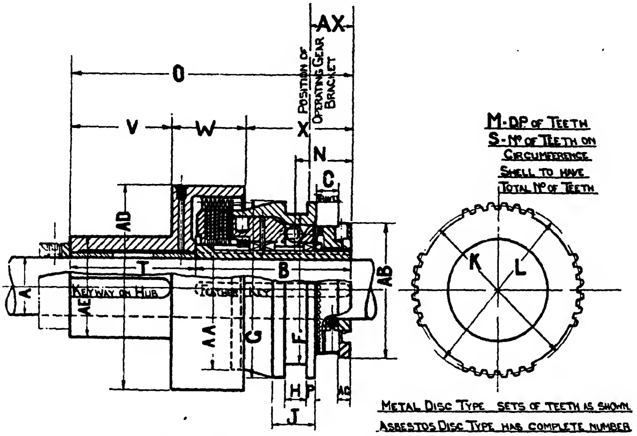

AX
O
POSITION OF OPERATING GEAR BRACKET
V
W
X
N
C
TRAVEL
AD
T
B
AB
A
AE
KEYWAY ON HUB
FEATHER KEY
AA
G
F
H
P
J
AG
M-DP OF TEETH
S-Nº OF TEETH ON CIRCUMFERENCE
SHELL TO HAVE TOTAL Nº OF TEETH
K
L
METAL DISC TYPE SETS OF TEETH AS SHOWN.
ASBESTOS DISC TYPE HAS COMPLETE NUMBER

Fig. 16.—EXTENDED HUB CLUTCH

POWER OUTPUTS AND APPROXIMATE DIMENSIONS IN INCHES

Size No. of Clutch	*H.P.* at 100 *r.p.m.*	*Max. Bore A*	*B*	*Travel C*	*F*	*G*	*H*	*J*	*K*	*L*
1	0·6	$1\frac{1}{8}$	$3\frac{21}{32}$	$\frac{9}{16}$	3	$3\frac{1}{2}$	$\frac{1}{2}$	1	3·20	3·366
2	1·0	$1\frac{1}{4}$	$3\frac{21}{32}$	$\frac{9}{16}$	$3\frac{1}{4}$	$3\frac{7}{8}$	$\frac{1}{2}$	1	3·60	3·766
3	2·0	$1\frac{1}{2}$	$4\frac{21}{32}$	$\frac{11}{16}$	$3\frac{7}{8}$	$4\frac{5}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	4·80	4·966
4	3·0	$1\frac{3}{4}$	$4\frac{21}{32}$	$\frac{11}{16}$	$4\frac{3}{8}$	$5\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	5·00	5·20
5	5·0	$2\frac{1}{4}$	5	$\frac{3}{4}$	$5\frac{1}{4}$	6	$\frac{11}{16}$	$1\frac{11}{32}$	6·00	6·20
6	7·5	$2\frac{7}{8}$	$5\frac{5}{16}$	$\frac{13}{16}$	6	7	$\frac{3}{4}$	$1\frac{7}{16}$	7·00	7·25
7	11·0	$3\frac{1}{2}$	$6\frac{1}{8}$	1	$7\frac{1}{4}$	$8\frac{1}{4}$	$\frac{7}{8}$	$1\frac{3}{4}$	8·40	8·686

Size No. of Clutch	*H.P. at* 100 *r.p.m.*	*M*	*N*	*O*	*P*	*S*	*T*	*V*	*W*	*X*
1	0·6	10/12	$1\frac{7}{16}$	$6\frac{9}{32}$	$\frac{1}{4}$	32	$2\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{11}{16}$	$2\frac{15}{32}$
2	1·0	10/12	$1\frac{7}{16}$	$6\frac{21}{32}$	$\frac{1}{4}$	36	3	$2\frac{1}{2}$	$1\frac{11}{16}$	$2\frac{15}{32}$
3	2·0	10/12	$1\frac{5}{8}$	$8\frac{9}{32}$	$\frac{1}{4}$	48	$3\frac{5}{8}$	3	$2\frac{1}{16}$	$3\frac{7}{32}$
4	3·0	8/10	$1\frac{5}{8}$	$8\frac{13}{32}$	$\frac{1}{4}$	40	$3\frac{3}{4}$	$3\frac{1}{8}$	$2\frac{1}{16}$	$3\frac{7}{32}$
5	5·0	8/10	$1\frac{13}{16}$	$9\frac{3}{4}$	$\frac{9}{32}$	48	$4\frac{3}{4}$	4	$2\frac{1}{4}$	$3\frac{1}{2}$
6	7·5	6/8	$1\frac{15}{16}$	$11\frac{5}{16}$	$\frac{5}{16}$	42	6	$5\frac{1}{4}$	$2\frac{1}{4}$	$3\frac{13}{16}$
7	11·0	5/7	$2\frac{5}{16}$	$13\frac{3}{8}$	$\frac{3}{8}$	42	$7\frac{1}{4}$	$6\frac{1}{2}$	$2\frac{1}{2}$	$4\frac{3}{8}$

Size No. of Clutch	*H.P. at* 100 *r.p.m.*	*AA*	*AB*	*AD*	*AE*	*AG*	*AX*	*Maximum Featherway for Maximum Bore*
1	0·6	3	$2\frac{9}{16}$	$3\frac{7}{8}$	2	$\frac{3}{8}$	$1\frac{5}{32}$	$\frac{5}{16} \times \frac{5}{32}$
2	1·0	$3\frac{1}{4}$	$2\frac{13}{16}$	$4\frac{3}{8}$	$2\frac{1}{8}$	$\frac{3}{8}$	$1\frac{5}{32}$	$\frac{5}{16} \times \frac{5}{32}$
3	2·0	$4\frac{1}{8}$	$3\frac{1}{2}$	$5\frac{5}{8}$	$2\frac{5}{8}$	$\frac{3}{8}$	$1\frac{9}{32}$	$\frac{3}{8} \times \frac{3}{16}$
4	3·0	$4\frac{9}{16}$	$3\frac{7}{8}$	$5\frac{13}{16}$	$2\frac{7}{8}$	$\frac{3}{8}$	$1\frac{9}{32}$	$\frac{3}{8} \times \frac{3}{16}$
5	5·0	$5\frac{7}{16}$	$4\frac{11}{16}$	$6\frac{7}{8}$	$3\frac{1}{2}$	$\frac{7}{16}$	$1\frac{7}{16}$	$\frac{1}{2} \times \frac{1}{4}$
6	7·5	$6\frac{3}{8}$	$5\frac{7}{16}$	8	$4\frac{1}{2}$	$\frac{7}{16}$	$1\frac{17}{32}$	$\frac{5}{8} \times \frac{5}{16}$
7	11·0	$7\frac{9}{16}$	$6\frac{9}{16}$	$9\frac{1}{2}$	$5\frac{1}{2}$	$\frac{1}{2}$	$1\frac{13}{16}$	$\frac{3}{4} \times \frac{3}{8}$

possible some form of flexible coupling must be embodied (Fig. 18).

Friction clutches can be combined with belt or rope pulleys, gear- or chain-wheels.

The actual method of attachment varies, of course, with the size and type of clutch, size of pulley, gear-wheel, etc., also with the type of bushing.

Upon the selection and attention given to the bushing in the loose portion, the success or otherwise of the installation depends.

Some alternative methods are shown in Figs. 19, 20, 21, 22.

One point to be carefully watched is the position of the bearing surface of the gunmetal bushes relative to the belt pull; an endeavour should be made to arrange this as near as possible central with the pull.

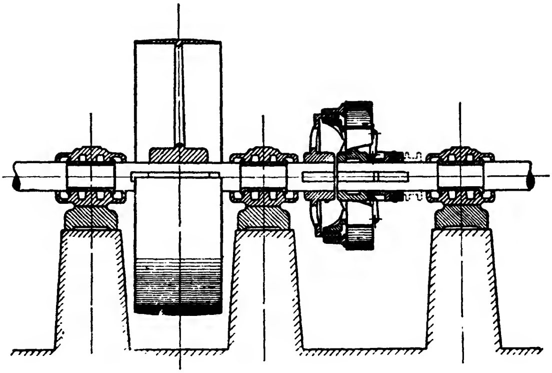


Fig. 17.—Rim type clutch coupling
Showing clutch rim keyed to extension shaft.

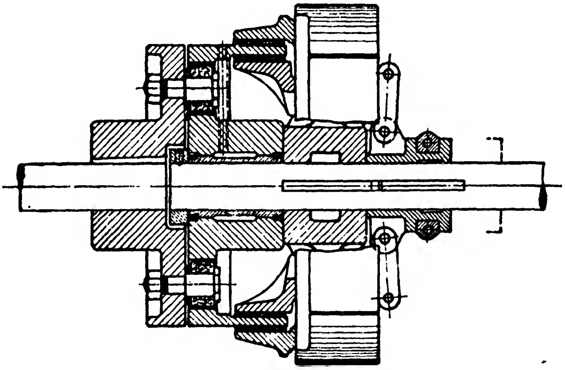


Fig. 18.—Arrangement showing rim clutch coupling combined with crown pin type flexible coupling

Lubrication

Oil may be used if desired, but it should not be fed direct on to the shaft, otherwise it soon runs away. A better arrangement is to feed into a reservoir

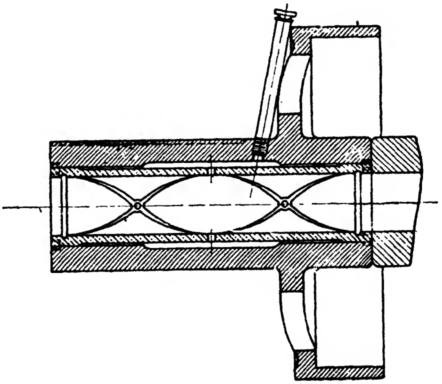


Fig. 19 (*right*).—Standard arrangement of whole (single) gunmetal bush to extended hub to rim clutch

Showing lubrication pipe and oil reservoir.

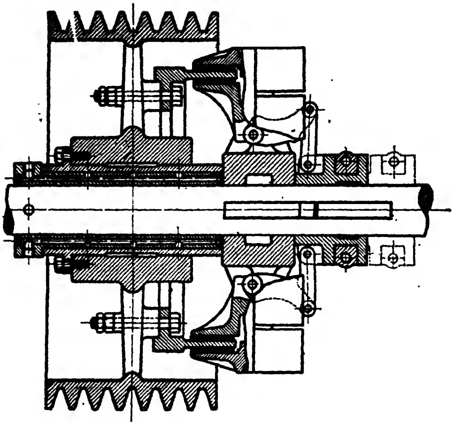


Fig. 20.—Rim type clutch pulley arrangement

Showing clutch rim bolted to a cast-iron V-rope pulley mounted upon split renewable gunmetal bush running on a cast-iron sleeve.

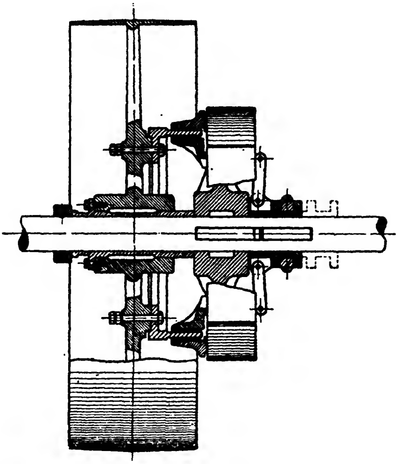


Fig. 21.—Rim type clutch pulley arrangement

Showing clutch rim bolted to cast-iron pulley mounted on a split renewable gunmetal bush running direct to shaft.

around the gunmetal bush and from this to the shaft through holes and grooves, as shown in Fig. 19.

As an alternative to oil, grease may be used, but it should be fed direct on to the bearing surface, and only good-quality grease, which does not solidify, should be used. Neglect of this point results in the grease passages becoming clogged up, with disastrous results to the bushing and shaft.

If the pulley portion is to revolve for other than short periods around a stationary shaft, it is worth while considering the application of a cast-iron

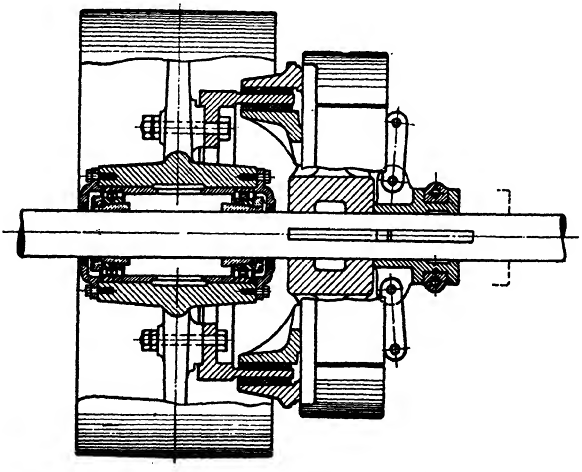


Fig. 22.—Shown fully mounted on ball bearings

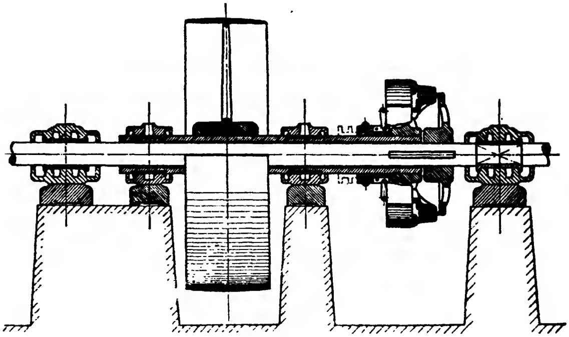


Fig. 23.—ARRANGEMENT OF QUILL DRIVE

Showing clutch mechanism keyed to quill and rim keyed to main shaft.

sleeve drilled and fitted with Stauffer lubricators, as shown in Fig. 20. This will allow lubrication to be given whilst the pulley is revolving, and also prevent damage to the shaft in the event of neglect of lubrication.

Never allow a pulley fitted with ordinary gunmetal bushes (Fig. 21) to stand on a revolving shaft for other than a very short time, otherwise excessive wear is bound to occur, and the shorter the bush the sooner will trouble begin.

Various means are adopted to overcome this difficulty, but for shafts up to, say, 5-in. diameter the introduction of ball or roller bearings, or a combination of ball and roller bearings, is the surest way of eliminating lubrication difficulties (Fig. 22).

For shafts larger than 5 in., or where it is not convenient to install ball bearings, a cast-iron or steel quill may be used, supported on independent bearings (Fig. 23).

From the illustrations it will be seen that the pulley is carried clear of the shaft, and if arranged so that the mechanism is at rest when the clutch is disengaged no lubrication whatever will be required when the clutch is disengaged, and no wear will be taking place in any part. This is the arrangement usually adopted when conditions demand that the clutch shall be disengaged for months at a time.

It is, of course, absolutely essential to see that the quill is set concentric with the shaft, and this point should be checked and corrected from time to time, in case wear in the quill bearings takes place, or the foundation settles.

OPERATING GEARS

As stated earlier in this section, a friction clutch should be so arranged as to give complete control of power. This, of course, requires an operating gear to be installed in such a manner that the operator can put the clutch in and out of operation without difficulty.

For small clutches ordinary hand-levers are quite suitable if arranged so that the weight of the lever does not tend to put the clutch into engagement—a simple point, but often overlooked.

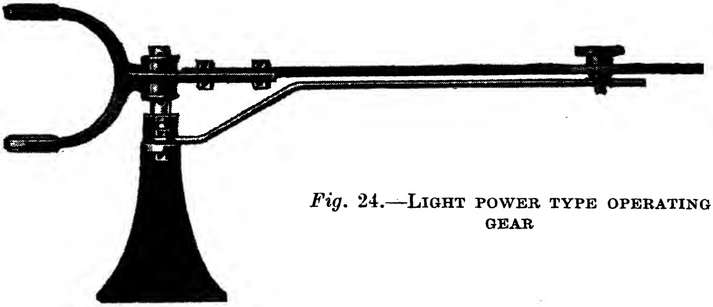


Fig. 24.—LIGHT POWER TYPE OPERATING GEAR

Screw- or Rack-type Gear

For clutches to transmit, say, 50 h.p. at 100 r.p.m., it is usually preferable to install a screw- or rack-type gear.

The actual form which these gears take depends entirely upon :

(*a*) The surroundings.

(*b*) The frequency of operation.

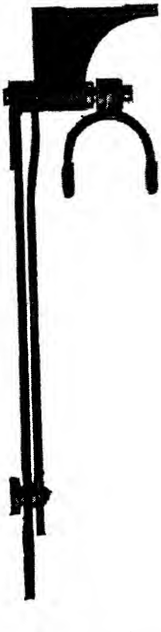


Fig. 25.—CEILING TYPE HAND-LEVER OPERATING GEAR

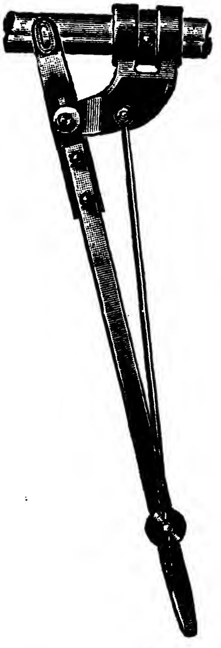


Fig. 26.—SELF-CONTAINED PANEL LEVER

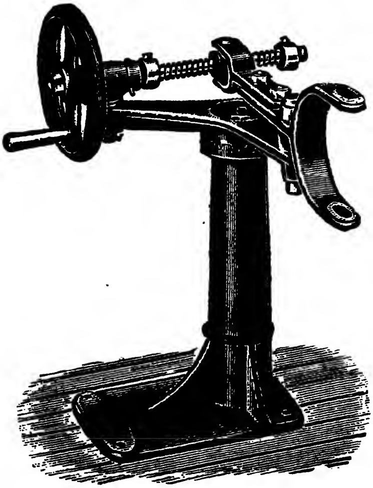


Fig. 27.—STANDARD SCREW TYPE OPERATING GEAR

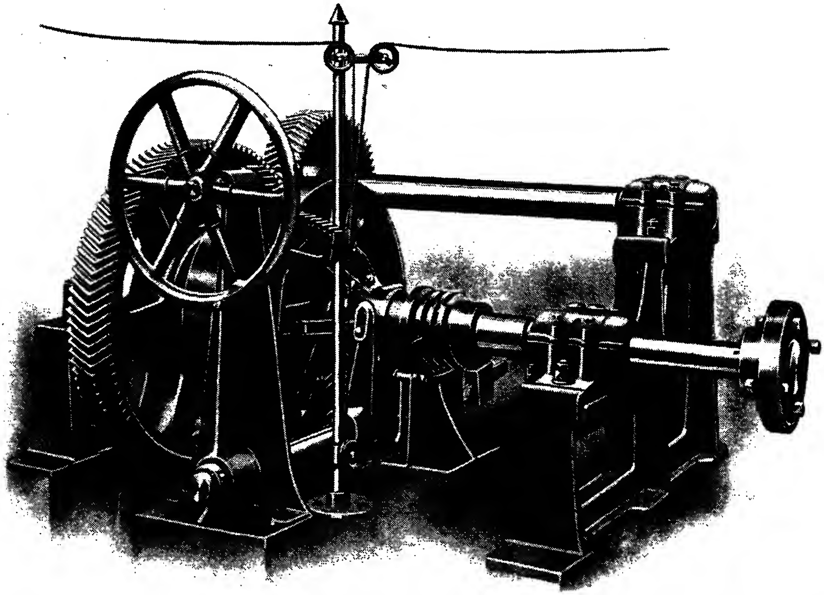


Fig. 28.—RIM TYPE CLUTCH INSTALLATION WITH INSTANTANEOUS CLUTCH THROWOUT GEAR

(*c*) Whether the clutch must be instantly operated in case of emergency.

In any installation it is an advantage to install a gear which is easily capable of putting the clutch in and out of operation without the operator having to exert the whole of his strength.

If a clutch is too large to be easily put in and out of operation by a simple lever, it is much more satisfactory to employ other mechanical means, say, screw- or gear-wheels, so that the clutch may be engaged and disengaged at the correct speed, having due regard to the work in hand.

Emergency Stopping

It is sometimes necessary to bring machines to rest instantly, in order to prevent injury to the operator, and Fig. 28 shows how it is possible to do this by means of a friction-clutch drive, so arranged that a slight pull on a wire rope is sufficient to disengage the clutch and so cut off the driving power. If desired, a brake can be incorporated.

From the illustrations it will be seen that operating gears may take a variety of forms, and offer scope for the draughtsman's ability to design a simple combination of levers or other mechanical means of increasing the actual pull given by the operator, so that the clutch can be put in and out of operation without undue effort from the most convenient position.

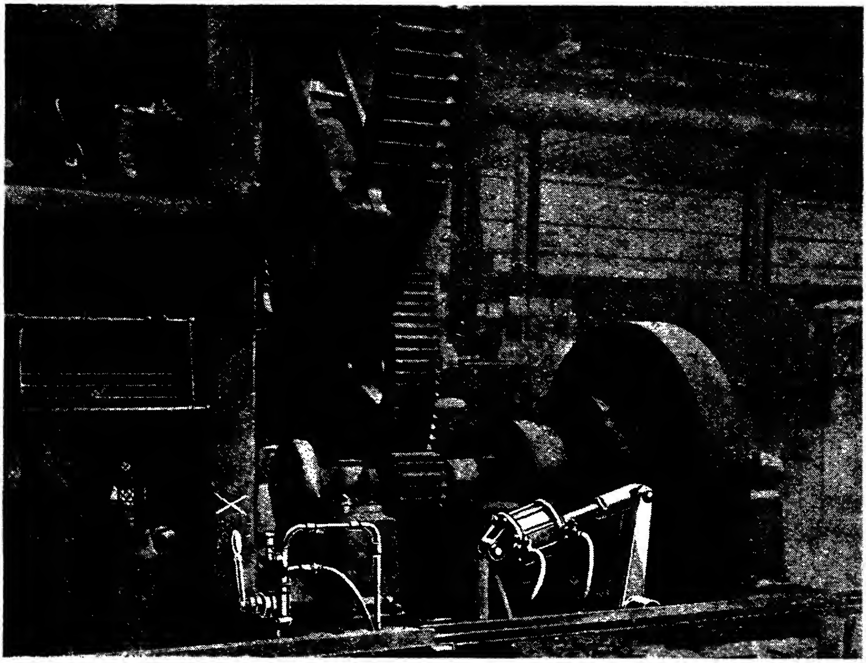


Fig. 29.—Air-operated friction clutch drive

Position of the Starting Lever

In any machine the starting lever of the clutch should be in such a position that the attendant can put the clutch in, or throw it out of, engagement from the position in which he stands when working the machine. Attention to this point will amply repay a little extra initial cost.

In arranging operating gears it should be remembered that long shafts spring a considerable amount, even when properly supported by rigid supports at short intervals ; such shafts should be kept as short and stiff as possible. Neglect of this often results in an excessive effort being required to operate the clutch. If levers and connecting rod can be used in place of a shaft, this should be considered.

When the point from which a clutch has to be operated is beyond the reach of shafts, connecting rods, etc., there is a temptation to arrange the operating levers, etc., where the starting operator has not a good view. There is then a possibility of the machine being started when all is not clear, and causing an accident. This is not likely to occur when the starting operator is near his work and can see what he is doing.

Operation by Compressed Air

If compressed air is available, a cylinder can be connected in such a way that a small handle control will operate a mechanical clutch of any size, and such an arrangement is shown used in connection with a large press, and controlled by the small lever x (Fig. 29).

Operation by Electricity

Where electric current is available (either A.C. or D.C.) it is possible to utilise a solenoid or small electric motor to operate a mechanical clutch, and Fig. 30 shows an arrangement using an electric motor.

It will be seen from the illustration that it is possible to put the clutch in and out of operation from any desired point, by pressing push-button-type switches, or if desirable to put the clutch in and out of gear by a master switch at any point, and out of gear only by push-button-type switches from any number of points; or, if the clutch is required to comply with the new Factory Act, the gear can be arranged so that the clutch can be put in gear by hand-wheel, and out of operation instantly by pressing a push-button-type switch arranged in any convenient position.

This arrangement is also used in connection with float-type switches, etc.

Fig. 30.—Electrically controlled operating gear with push-button switch

EMERGENCY SLIPPING DEVICES

It is, of course, possible to design clutches so that they will slip if called upon to transmit any serious overload, thus preventing breakdowns, etc. This is a point which should always be kept in mind.

There are, however, many installations where it is not necessary to

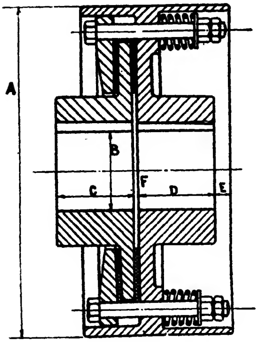


Fig. 31.—EMERGENCY SLIPPING DEVICE

have an operated friction clutch, but where it would be of advantage if a device could be fitted to slip when a serious overload occurs.

For such installations, slipping devices are used as illustrated in Fig. 31, and they are made in a variety of forms.

They are designed to protect machinery, electric motors, etc., from breakdown, due to overloads, shock loads, etc., and are of simple construction, cheap to install, can be set to slip at any specific overload, and once correctly set require no further adjustment.

Driving power is governed by pressure applied through bolts and springs, the contact faces being lined with friction material.

These devices are rapidly taking the place of couplings fitted with shear pins ; they are most efficient, and prevent the undue waste of time which occurs with such couplings after pin has sheared due to overload, fatigue, etc.

LIST OF POWERS AND DIMENSIONS FOR SLIPPING COUPLINGS

Outside diameter . . A	*in.* 4¾	*in.* 6	*in.* 7½	*in.* 9	*in.* 10½	*in.* 12
Maximum bore . . B	1	1½	1¾	2	2½	3
Maximum bore . . C	1⅝	1¾	2	2¼	2¾	3¼
Maximum bore . . D	1⅝	1¾	2	2¼	2¾	3¼
Maximum bore . . E	⅜	7/16	½	½	⅜	¼
Maximum bore . . F	⅛	⅛	⅛	⅛	⅛	⅛
Maximum H.P. at 100 r.p.m.	·25	·75	1·75	3·25	4·5	8

Outside diameter . . A	*in.* 14	*in.* 16	*in.* 18	*in.* 21	*in.* 24	*in.* 27
Maximum bore . . B	3½	4	4½	5	6	7
Maximum bore . . C	3¾	4¼	5	6	7	8
Maximum bore . . D	3¾	4¼	5	6	7	8
Maximum bore . . E	⅛	⅛	—	—	—	—
Maximum bore . . F	⅛	3/16	3/16	¼	¼	¼
Maximum H.P. at 100 r.p.m.	12	18	28	42	60	80

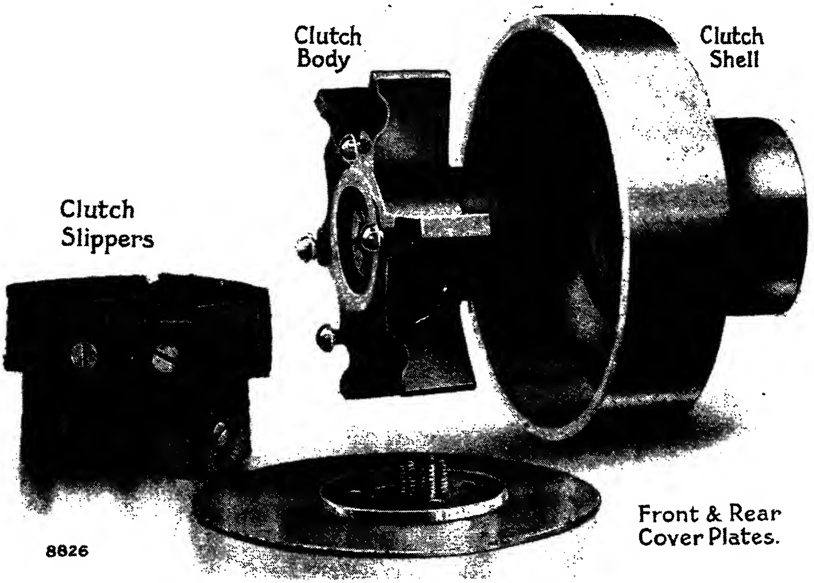


Fig. 32.—The component parts of a centrifugal clutch

They may be made either as couplings for connecting two shafts, or the slipping device may be provided with an extended boss for keying on pulley, gear-wheel, chain sprocket, etc. Multiple-plate design may be used for heavy drives and high speeds.

CENTRIFUGAL CLUTCHES

We have seen how it is possible to control power from a large prime mover, and will now consider the electric motor drive which is so popular, and which is being installed in order to avoid the use of shafting.

Where motors have to drive machinery which does not require more than about one-third full-load torque to start, there is no difficulty; but if the starting torque is heavy, special consideration must be given, otherwise the motor may not be able to attain full speed.

It is true that the installation of suitable motors and switch-gear will get over the difficulty, but in many cases friction clutches would enable smaller and cheaper motors and cheaper switch-gear to be used.

The squirrel-cage motor is recognised as a simple and reliable type of motor, but the ordinary design labours under the serious disadvantage that when starting against full load, about six times full-load current is required, and many supply authorities impose restrictions to avoid this sudden demand for current.

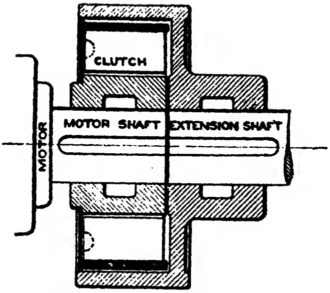


Fig. 33.—Type "A" centrifugal clutch with shaft coupling

If started by star-delta starter, the voltage on the star side is reduced and the current limited. The motor then gives a starting effort of about one-third full-load torque, which is not sufficient to get many industrial drives up to speed.

A transformer-type starter can be used, which will give about 70 per cent. full-load torque with three times full-load current, but if this sudden demand for current will not be allowed, and a clutch is not fitted, a slip-ring motor must be installed.

From the above remarks it will be evident that, if it is possible to relieve the motor of load until it has attained full speed, starting difficulties will disappear. On heavy drives this is done by the application of operated friction clutches either coupled direct to the motor or placed on the driven shaft, so that the motor is allowed to attain its full speed without being called upon to do any work, and is connected to the driven member when running under the most favourable conditions.

In cases where the starting torque required does not exceed, say, three-quarters full-load torque, it is not necessary to install an operated clutch, and for these conditions automatic clutches operated by centrifugal force are installed on the motor shaft. These are made in a variety of designs, and enable ordinary squirrel-cage motors with star-delta starters to be used for ordinary industrial drives which would otherwise call for slip-ring motors.

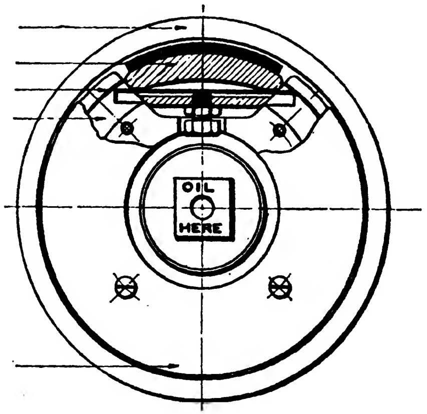


Fig. 34.—Elevation and section through flat spring, showing method of adjustment of driving pressure

Description and Use

They facilitate the use of inexpensive motors and simple switchgear, and assist prime movers to overcome starting difficulties by relieving them of the large initial effort usually required to set the load in motion. Thus, by allowing the load to be picked up in a steady and progressive manner, they considerably reduce the peak loads at starting and are of great assistance to prime movers and motors of low starting torque.

Centrifugal clutches are of two types, i.e. with and without spring control. Where motors are required to start up against more than 75 per cent. of full load the spring-control type is recommended. With either type, engagement is automatic and gradual, relieving both prime mover and driven machine from starting shocks. There are no complicated mechanical parts. Slippers are lined with asbestos friction surfaces having an exceptionally long life.

If fitted with spring control, the speed at which engagement takes place can be predetermined.

Overloads are taken care of by this clutch. The slippers can be regulated to transmit a specified load and to slip when this load is exceeded, the clutch automatically picking up the load again when speed is reduced to normal.

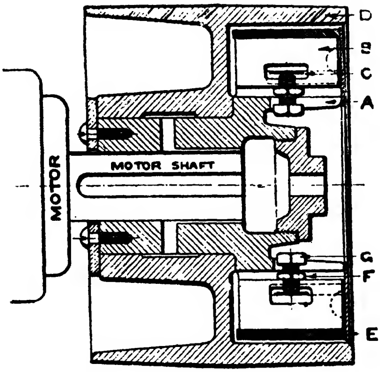


Fig. 35.—SECTIONAL VIEW WITH SPRING CONTROL

Given ordinary care in lubrication, centrifugal clutches require no further attention.

Construction

Centrifugal clutches consist of three main components, body slippers (which form the driving portion) and shell, or driven part.

The body is keyed to the driving shaft and has a number of pockets into which are fitted the slippers.

Slippers are faced on the outside with asbestos friction linings, which come into contact with inside of clutch shell when clutch is engaged.

The clutch shell or driven member is either fixed to the driven shaft (as in coupling clutch), or is constructed to run loose on the hub of the clutch body and also form, or carry pulley, gear-wheel, or chain sprocket, etc.

As the body rotates, the slippers are thrown outwards by centrifugal

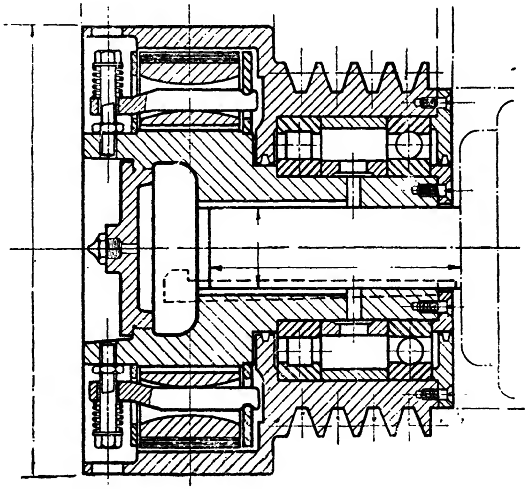


Fig. 36.—CENTRIFUGAL CLUTCH WITH PULLEY PORTION MOUNTED ON BALL BEARINGS force, and the asbestos facings press on the inside of the clutch shell with growing pressure as velocity increases, until full speed is attained; the weight of the slippers being sufficient to transmit up to the maximum horse-power stipulated.

The action is gradual, steady, and automatic, and as the clutches are constructed with minimum parts, and no links, pins, toggles, etc., are used, trouble is negligible.

Important.—Centrifugal clutches are designed for loads the starting up of which occupies periods up to 30 seconds; with longer periods heat is rapidly generated, and if unduly extended becomes a source of weakness and danger.

SPRING-CONTROL TYPE

In this type, springs prevent the slippers from making contact with the shell until the motor has accelerated to 75 per cent. of full speed, or any lower predetermined speed to suit requirements.

Spring-controlled clutches are necessary in all cases when used with single-phase motors and D.C. shunt-wound motors.

Up to 50 h.p. it is usually found that 2- and 3-phase squirrel-cage motors, and compound-wound D.C. motors when fitted with centrifugal clutches without springs, are quite suitable for ordinary installations, but above 50 h.p. and also with single-phase motors and D.C. shunt-wound motors, spring-controlled centrifugal clutches are necessary.

In some instances single-phase motors are quite unable to start if called upon to overcome any load whatever, and it is advisable to fit centrifugal clutches, the pulley portions of which are mounted on ball bearings (Fig. 36). This illustration indicates a patented design of clutch made by Messrs. Crofts (Engineers) Ltd., and it will be noted that adjustment of driving pressure can be made without removing cover, plates, slippers, etc., as is necessary with other designs.

Friction clutches can now be used with every confidence, and if they are correctly installed they will give good service. If lines of shafting are run for long periods when not required, a friction clutch to enable them to be brought to rest will soon pay for itself in power saved.

Chapter V

THE INSTALLATION OF MAGNETIC CLUTCHES

THE magnetic clutch is the easiest type both to install and to maintain, because of the absence of complicated external operating gear. The majority of magnetic clutches operate by the clamping effect produced by a rotating-magnet system. There are some exceptions, and these will be mentioned later.

SINGLE-PLATE CLUTCH

Method of Operation

The most common type of magnetic clutch is the single-plate unit shown in Fig. 1, from which the method of operation should be quite clear. A magnetic field is set up by the rotating magnet carrying the magnetising coil, the armature is attracted towards the body and is prevented from making contact with it by the projection of the friction linings, and the pressure of this armature on to the linings provides the means of transmitting the torque.

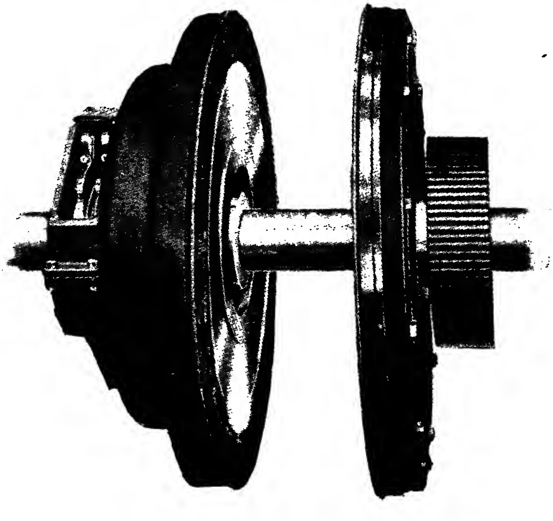


Fig. 1.—SECTIONAL VIEW OF SINGLE-PLATE CLUTCH

The installation of the clutch will depend on the shaft arrangement, which may be either a through shaft carrying both parts or two independent shafts, the most common arrangement being the latter.

Mounting Clutch Body on Shaft

Mounting of the clutch body on to its shaft calls for care if satisfactory performance is to be obtained. The bore of the body

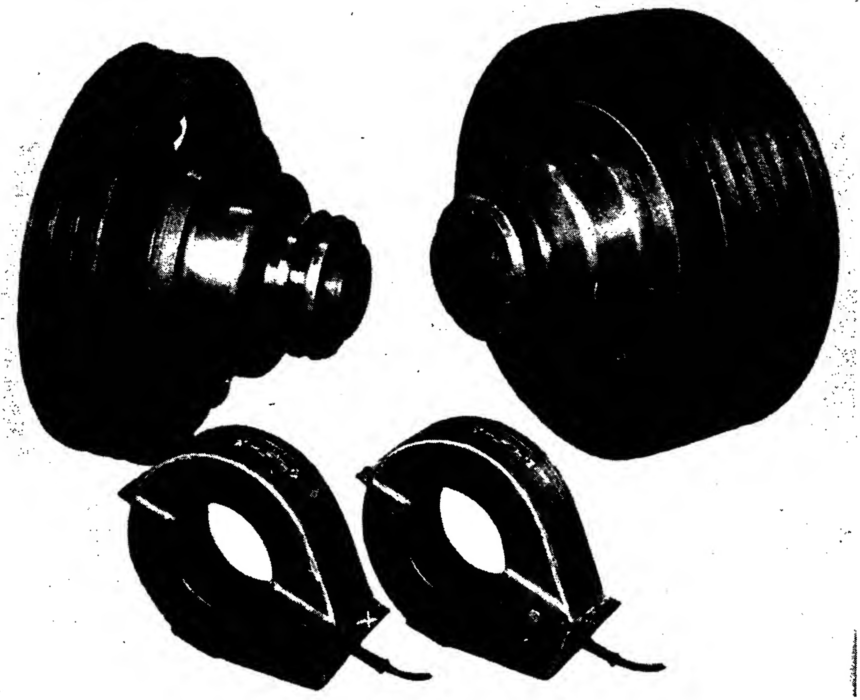


Fig. 2.—SINGLE-PLATE CLUTCH FOR THROUGH SHAFT

will be arranged according to the method of pressing adopted. In the case of the use of a hydraulic press, the bore will be slightly smaller than when it is to be forced on by clamps. The key will be placed in position before pressing on to ensure proper alignment, and care must be taken that pressure is applied in perfect alignment with the axis of the shaft to prevent any distortion of the body, as it is essential that the friction-lining face be perfectly square with the shaft.

In the case of clutches only lightly pressed on, a taper key should be used, so that any tendency for the body to be pulled forward to the armature is prevented by the fact that any movement tightens the body more on to the key.

Clutches embodying a through shaft are arranged for the armature to run on bearings supported on a sleeve usually attached to the clutch body. This method enables a suitable thrust race to be incorporated on the sleeve. A typical arrangement of this type is shown in Fig. 2. In this case, ball bearings are incorporated. Erection is the same if gunmetal bearings are used. As there is no tendency for the unit to move on its shaft, it is not necessary to key so tightly.

When clutches incorporate spring plates as shown in Fig. 1 and are mounted on separate shafts, provision must be made to carry the end-thrust load set up when the clutch is energised and the plate deflected. This load value varies according to the diameter of the clutch, and unless the shaft bearings are capable of withstanding this load a suitable thrust bearing, usually a gunmetal collar, will be attached to the magnet body and located so that the armature hub will rub against it, provision being made for lubrication.

Checking Alignment of Clutch

Alignment of the two halves of the clutch should be checked after erection, except in the case of an armature mounted on a self-contained sleeve. The two halves of the clutch, being machined all over, a straight-edge may be laid against the outer edges and checked. A further check may be made by gauging the air-gap clearance at different diametrically opposite points.

Air-gap Measurement

The air-gap measurement is the most important to be watched when the bodies are mounted on separate shafts ; should this gap be too great, then it is possible that the clutch will not transmit its full torque.

Taking up Wear in Clutch

Maintenance is a simple matter, as the only main point requiring attention is the air gap, which will increase as the linings wear down. Actually, this wear is usually very slow, as the normal pressure is low. The method of taking up the wear will depend on the design of the clutch. In some cases special screws, termed jacking screws, are provided ; in others, shims are used. It is usual for the linings to be riveted on to steel supporting rings, and in the first-mentioned case the bolts clamping the linings would be slackened off and the jacking screws moved forward until the gap was the required dimension and even all around the circumference, after which the locknuts would be tightened up, and then the ring-holding bolts would be clamped down tight and locked.

The actual air-gap dimension will vary according to the clutch size and make, and normally varies from $\frac{1}{32}$ in. to $\frac{1}{4}$ in., the amount of wear allowed being 50 per cent. to 100 per cent. of the air-gap original dimension. The actual amount of wear allowed before relining is required is dependent on its original thickness, as wear must not take place on the actual rivets to prevent scoring of the armature face. Usually, wearing away of the lining to half or a third of its original thickness is permissible.

Relining the Clutch

Relining is a simple matter, and the original rings should be used as a template for marking out the rivet holes of the new lining. Two types of rivets are employed, the taper-head aluminium being very popular, but for some rigid-moulded linings it is essential to employ a flat-headed rivet to prevent splitting of the lining by the taper head.

The same grade of lining material should be used for relining unless the performance of the original has not given every satisfaction. This point is very important, as the temperature limits for the different grades vary very widely, and if a fibre lining was used to replace an asbestos grade for heavy work, it would be rapidly destroyed.

Remedying Clutch Slip

Sometimes trouble is caused by oil and grease getting on to the linings and causing slipping. This can be removed by washing with petrol, and under no circumstances should it be burnt off, as this is liable to destroy the lining.

Bearings such as have been mentioned mounted in hubs or the thrust collar between the two halves of the clutch should be lubricated with a good grease at regular intervals, such as would be normally given to other plant having that type of bearing.

Attention to Brush Gear Slip Rings

Attention must be given to the brush gear and slip rings, as if this point be neglected and bad contact be made, the excessive voltage drop may reduce the magnetising current sufficiently to affect the torque transmitted by the clutch. Routine attention should be given to the surfaces of the rings, which are usually of gunmetal, and to the brush face, so as to ensure even contact all over.

The size of brush usually employed is far bigger than actually required if based on current density, but a good-sized brush is required to ensure reliability. The brush grade is selected according to the speed and current rating of the coil ; every effort should be made to replace by the same grade if at all possible.

The brush tension should be adjusted according to the wear ; this is done either by providing an adjusting screw on the spring or, alternatively, the spring tension is constant and adjustment is made by moving round the complete brushholder on its insulated bar.

MULTIPLE-PLATE CLUTCH

The multiple-plate clutch is not so common as the single-plate, due to its much higher cost, but is invaluable for the control of loads of very high inertia, due to the very large lining area which is accommodated, enabling heavy starts to be made repeatedly without the risk of any overheating.

An exploded view of this type of clutch is shown in Fig. 3. Here can be seen the plates carrying the friction linings, and the keyed hub on which they slide, this latter hub being mounted on the driven shaft.

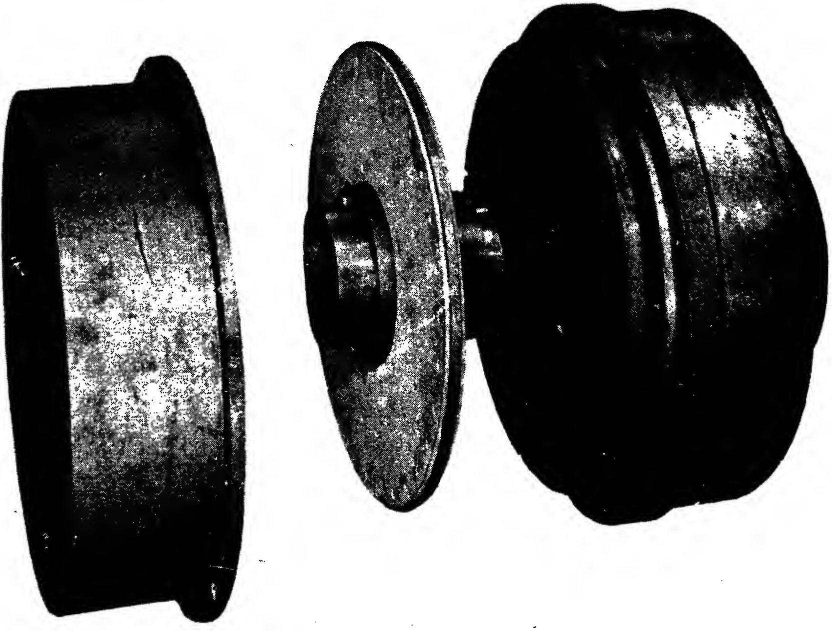


Fig. 3.—VIEW OF MULTI-PLATE CLUTCH DISMANTLED

All the other parts of the clutch are mounted on the driving shaft. The complete magnet arrangement with armature is shown on the right-hand side of the photograph. The outer case in which the rubbing plates slide can also be seen.

An important feature of this type of clutch is the fact that little weight is carried on the driven shaft, so that when the clutch is disconnected, there is little tendency to drift, the ratio of WR^2 of the driven and driving halves of the clutch will vary according to the size, but is of the order of 12 to 1.

Erection

The erection of this type will depend on the situation. The writer has erected one such unit weighing three-quarters of a ton over 10 ft. up and mounted between two big wall brackets, so that there was hardly any room to slide the parts sideways.

Normally, given reasonable conditions, the magnet body, complete with its hub, will be keyed to the driving shaft, then the hub of the driven side will be mounted on its shaft. The outer shell complete with its large adjusting ring will be threaded on to the driven shaft, followed by a friction lining, driving disc, and so on in this order. In a large unit there

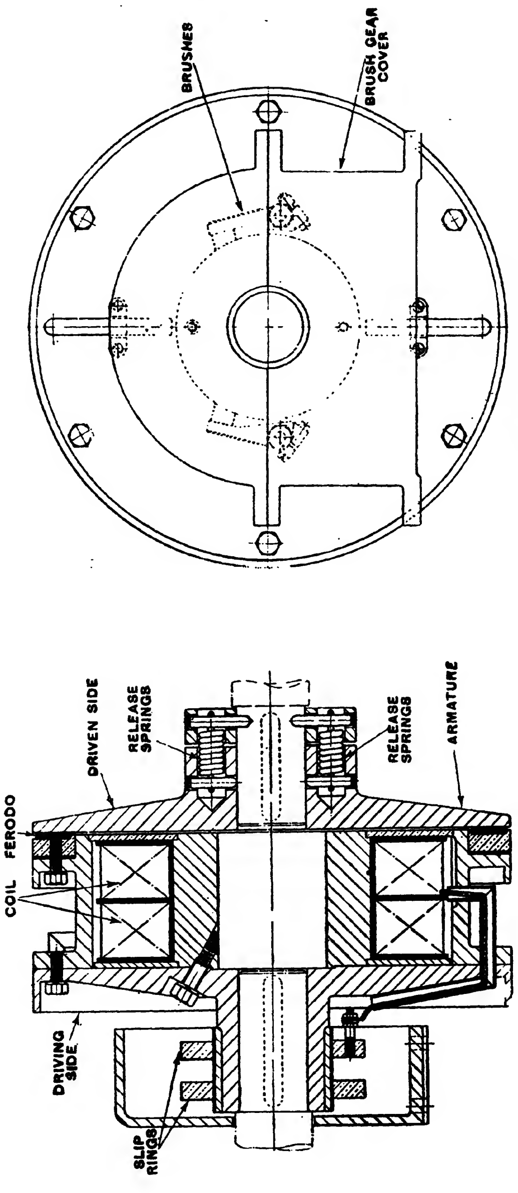

BRUSHES
BRUSH GEAR COVER
DRIVEN SIDE
RELEASE SPRINGS
RELEASE SPRINGS
ARMATURE
FERODO
COIL
DRIVING SIDE
SLIP RINGS

Fig. 4.—Sectional view of a typical Witton-Kramer magnetic clutch may be a total of twelve plates to assemble. According to the site conditions, the gunmetal thrust ring will be mounted in the magnet hub, suitably greased, and the labyrinth gland mounted up. The opportunity will be taken to grease fully while assembling.

The two sections must then be brought together with care. This is an operation which requires careful slinging to avoid damage to any part. The shaft has to be entered into the gland and thrust collar ; this, by the way, also serves as a bearing to keep the two shafts in line. When this operation is complete, the main body will be brought up and the circumferential bolts inserted, a check being made, before they are tightened up, that all is free and in line. Afterwards the bolts will be tightened up ; these bolts transmit the torque as well as holding the job together.

Adjustment

The pressure plate does not require any adjustment, as this is done at the works. Adjustment of pressure is made by the large adjustment ring mounted at the end of the clutch. When the approximate adjustment has been made, the magnet will be switched on and the gap between the armature and magnet face measured and compared with the figure given by the makers, usually about $\frac{3}{16}$ in. If this gap is exceeded, then the switch will be opened and the adjustment ring closed in until finally the correct adjustment is obtained. Providing the shaft alignment has been checked, the unit can be run.

Due to the fact that there are a considerable number of linings, a small amount of wear or bedding in will make an appreciable difference to the magnet gap. This point must be watched during the first dozen or so operations, so that there is no risk of the armature jamming hard against the magnet face.

The type of brush gear used may vary, and may be slip rings, as mentioned on the single-plate clutch, in which case the same maintenance will be adopted, or alternatively a plate type of slip rings, as shown in the figure, may be used. Here it is more usual to use a plunger type of brush, due to the limited amount of space available.

A lubricator will be found in a convenient position for greasing the thrust collar. A grease gun should be used preferably to ensure sufficient pressure to reach the actual bearing through the long pipe, and only a limited amount should be applied at one time to avoid the grease passing on to the friction linings. A high-melting-point grease should be used for this purpose.

Relining

The intervals between relining are usually very much longer than for any other type, but, naturally, this is a much bigger operation to carry out.

Firstly, the clutch must be opened out in the reverse manner to which it was assembled, and then when all the plates are exposed the old rivets will be knocked out. If circumstances permit, the old linings will be removed in a single circle; otherwise they can be cut into halves and the new linings inserted in a similar manner. Either the old linings or the steel rings can be used for a template. It is usual to use semi-tubular rivets, as a lining must be riveted to each side of the plate, and this type is easier to clinch up with a special tool.

TOOTHED-TYPE COUPLING

The toothed-type coupling is used for special applications, such as for coupling shafts, which must be kept in step and yet be capable of individual adjustment, a typical case being a screw-down gear on a rolling mill.

The general construction of this type of clutch is very similar to the single-plate unit, but the friction lining is missing, and in its place is cut a specially shaped toothed ring around the circumference. The angle of the teeth and spacing is chosen to avoid any tendency to sticking, and at the same time to transmit the maximum torque. The conventional armature is retained, but is arranged to slide on the hub instead of using a spring plate. A spring arrangement is incorporated to force the armature back when the magnet is switched off.

Maintenance

The maintenance of this type of clutch is very simple, and solely consists of keeping the toothed faces clean and the spring free in its case, so as to prevent sticking. Attention should also be given to the slip rings, as previously described. The question of relining does not apply in this case.

Erection

Erection is similar to the single-plate, but as the spring plate is omitted there is no need for a thrust bearing, with its attendant need for lubrication.

STATIONARY CLUTCH

A different type of clutch is the stationary type, as shown in Fig. 5. Here the magnet coil is stationary and operates through an air gap to pull together the two halves of the clutch. It will be noted that the two cones are both connected to the one shaft, the actual coupling together being done by the bolts carrying the springs, the inner cone being coupled to the second shaft. No friction linings are used, the connection being metal to metal.

It will be noted that the magnet system, being stationary, is not subject to mechanical stresses or problems of carrying the current to the coil. These couplings are used for the very highest speeds and powers. A new coupling of this type is now on the market, which permits a definite amount of misalignment.

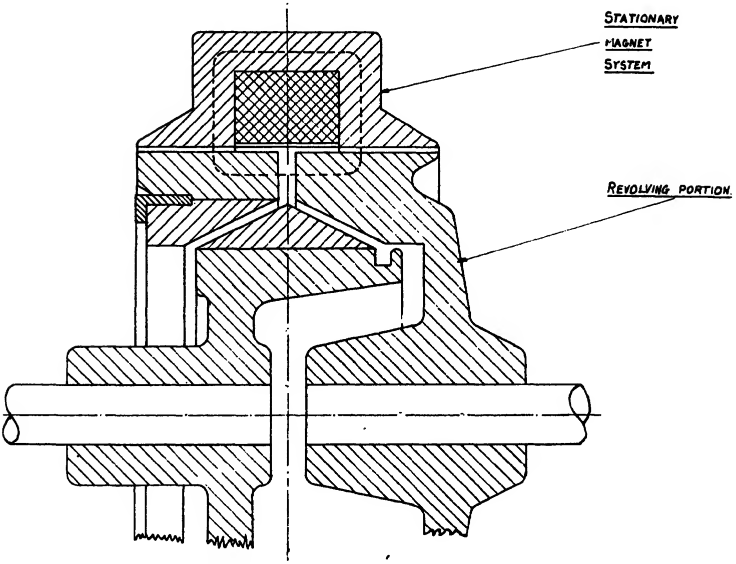


Fig. 5.—FORSTER STATIONARY-COIL MAGNETIC CLUTCH

Erection

The erection of this type follows somewhat on the lines of the multi-plate type previously mentioned ; the driving-cone section to be mounted on the one shaft and keyed on. The other section of the outer cone will be threaded over the driven shaft, and then the hub carrying the inner cone will be mounted on its shaft and the loose section of the outer cone brought up to the main section and the spring tension adjusted in accordance with the detailed instructions for the clutch size in question.

Provision is made for lubricating the friction faces, but should excessive wear take place, due to neglect of this operation, then provision is made for taking up wear by the adjustable cone section in the outer body, which is arranged to be adjustable relative to this piece.

The mounting of the magnet section is simple if a reasonable foundation is provided. The inner diameter must be truly concentric with the outer face of the conical section, and shims may be placed under the magnet feet to enable this to be accurately aligned.

Electrical Wiring of All Types of Clutches

The electrical wiring of all these clutches does not present any great difficulty. The most important point to be watched is that the voltage drop in the leads is kept at a minimum.

The usual attention should be given to the switch contacts, whether hand-operated or contactor type. A magnetic circuit of the types described is highly inductive, and a very vicious spark is given every time the switch is operated, and unless a proper discharge resistance is used, rapid destruction of the contacts takes place. Under no circumstances should the clutch be operated without the discharge resistance in circuit, unless the matter is of the greatest urgency, as there is a very considerable risk of breaking down the insulation of the magnet winding.

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